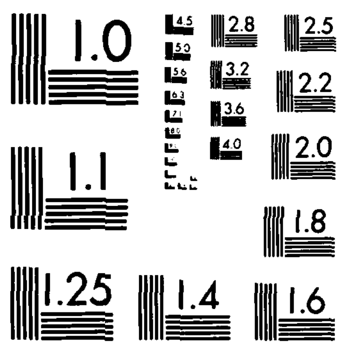
AD-A159 086 PROCEEDINGS OF THE SHIP CONTROL SYSTEMS SYMPOSIUM (5TH) 1/2
HELD AT U S NAVAL..(U) DAVID W TAYLOR NAVAL SHIP
RESEARCH AND DEVELOPMENT CENTER ANN.. E G BURT ET AL.
UNCLASSIFIED 03 NOV 78 F/G 13/10 NL



CROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

AD-A159 086

PROCEEDINGS

FIFTH SHIP CONTROL SYSTEMS SYMPOSIUM

OCTOBER 30 – NOVEMBER 3, 1978

VOLUME 6

-: PUBLICATION INFORMATION :-

These papers were printed just as received from the authors in order to assure their availability for the Symposium.

Statements and opinions contained therein are those of the authors and are not to be construed as official or reflecting the views of the Navy Department or of the naval service at large.

Any paper involved with copyrighting is prominently marked with the copyright symbol and was released for publication in these Proceedings.

Requests for information regarding the Proceedings, the Symposium, or the sponsor - David W. Taylor Naval Ship Research and Development Center - should be addressed to the Commander, David W. Taylor Naval Ship Research and Development Center, Bethesda, Maryland 20084 (Attn: Code 273).

VOLUME 6

CONTENTS

UNDERWATER ELECTROMAGNETIC TRANSMISSION AND ITS APPLICATION TO SHIP POSITION CONTROL

by E.G.C. Burt and L. Rigby
Imperial College, London

ABSTRACT

The paper discusses the possible use of electromagnetic methods for the position-detecting element in a ship control system, where the need is to keep the ship over a selected point on the ocean floor in deep water - e.g. for sea bed drilling operations. Such systems currently employ acoustic methods for the determination of position; receivers attached to the ship's hull detect the arrival times of pulses from an ultrasonic transmitter on the ocean floor, the time differences giving a measure of the lateral positional error. The method however suffers from the effects of anomalous acoustic propagation, which can lead to gross errors in the estimate of position.

Electromagnetic waves in a conducting medium such as salt water are attenuated to a far greater extent than acoustic waves at all but the lowest frequencies; on the other hand, it is to be expected that at these long wavelengths the effects of small-scale irregularities in the medium will be far less pronounced. The transmission of both transient and sinusoidal E-M waves is investigated, and the paper also discusses the effects of the sea-air interface on wave propagation and diffusion. The results of some preliminary experiments are given, and their relevance to the ship control problem is briefly examined.

TRANSIENT FIELDS IN A CONDUCTING MEDIUM

Linear Source

We consider first the simplest possible E-M source: a current-carrying linear element of length $d\ell$. Expressions have been derived for the electric and magnetic field produced by such a source when the current is a δ-function; for example, the magnetic field is given by

$$
\begin{aligned}
H_\phi &= 0 , \quad t < \alpha \\
&= \frac{d\ell \sin\theta}{4\pi} \left\{ e^{-\alpha\beta} \left[\frac{\beta}{vr} \left(1 + \frac{\alpha\beta}{2} \right) + \frac{1}{r^2} \right] \delta(t-\alpha) \right. \\
&\qquad \left. + \frac{r\beta e^{-\beta t}}{v^3 (t^2 - \alpha^2)} \left[\beta I_o - \frac{2}{(t^2-\alpha^2)^{\frac{1}{2}}} I_1 \right] \right\} , \quad t \geq \alpha
\end{aligned} \tag{1}
$$

with similar expressions for E_r and E_θ ($H_r = H_\theta = E_\phi = 0$). The element $d\ell$ is at the origin of coordinates (r, θ, ϕ) and directed along the polar axis; the medium has

conductivity σ, permeability μ and permittivity ε, with $\beta = \frac{\sigma}{2\varepsilon}$, $\alpha = \frac{r}{v}$ and $v = \frac{1}{(\mu\varepsilon)^{\frac{1}{2}}}$; and the argument of the Bessel functions I_o, I_1 is $\beta(t^2-\alpha^2)^{\frac{1}{2}}$. These formulae are derived on the assumption that the medium is infinite in extent; so they only apply if both source and field points are clear of bounding surfaces between different media - e.g. the ocean floor or the sea surface (but see below).

It will be seen that the response at a distance r from the source is zero until $t = \frac{r}{v}$, at which time there is a δ impulse of H_ϕ, whose coefficient is however enormously attenuated with distance from the source (the term

$e^{-\alpha\beta} = \exp(- \frac{\sigma}{2\varepsilon v})r$, which for sea water becomes e^{-1} when r is a few centimetres). The main response is due to the term involving I_o and I_1 , propagating at a speed which decreases rapidly with distance.

This behaviour may be seen in Fig. 1, which shows the magnetic field due to a unit step of current, obtained by integrating equation (1):

$$h_\phi = \bar{r} \exp(- \bar{r} \sqrt{(1+\lambda^2)}) [I_o(\lambda\bar{r}) + I_1(\lambda\bar{r}) \sqrt{(1+ \frac{1}{\lambda^2})}]$$

$$+ 1 - \bar{r} \int_\lambda^\infty \frac{I_1(\nu\bar{r}) \exp(- \bar{r} \sqrt{(1+\nu^2)})}{\sqrt{(1+\nu^2)}} d\nu , \qquad \lambda \geq 0 \qquad (2)$$

where we have used normalised time $\bar{t}$ and range $\bar{r}$, given by

$\bar{r} = \frac{r}{r_o}$, $r_o = \frac{2}{\sigma} (\frac{\varepsilon}{\mu})^{\frac{1}{2}}$

$\bar{t} = \frac{t}{t_o}$, $t_o = \frac{2\varepsilon}{\sigma}$,

$\lambda = (\frac{\bar{t}^2}{\bar{r}^2} - 1)^{\frac{1}{2}}$

and

$$h_\phi = \frac{H_\phi(\bar{r},\lambda)}{H_\phi(\bar{r},\infty)} = \frac{H_\phi(\bar{r},\lambda)}{d\ell \sin\theta/4\pi r^2}$$

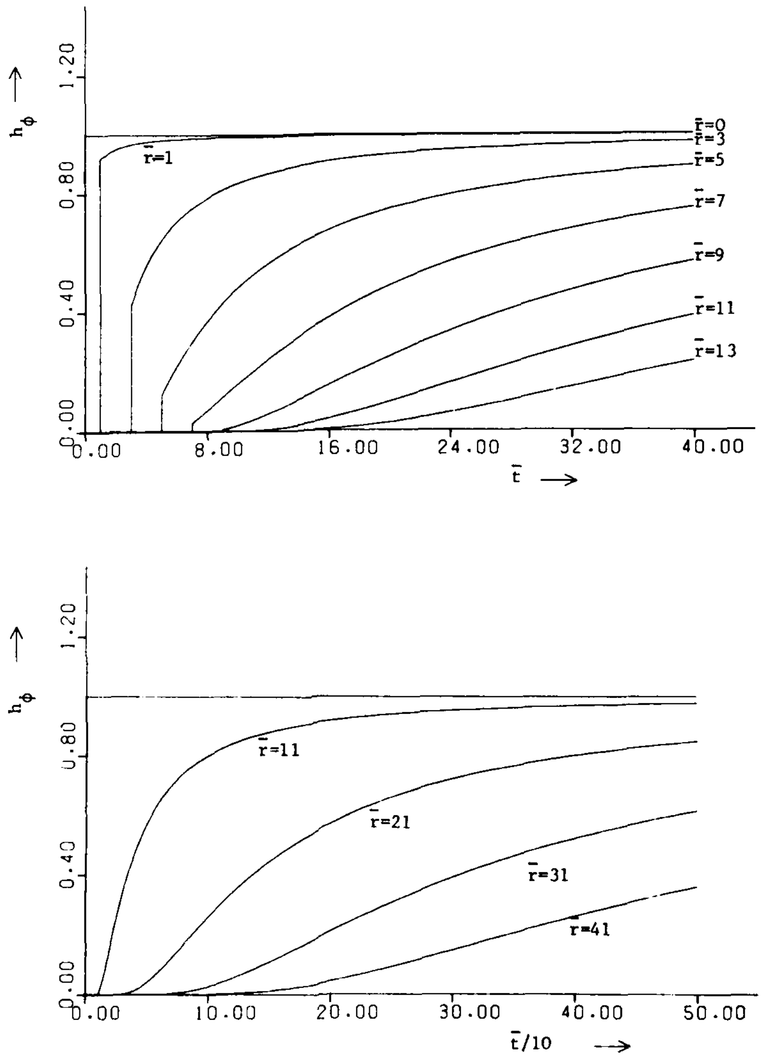


Figure 1. Magnetic Field due to a Unit Step of Current in an Element $d\ell$

It will be observed that, at short ranges, the response approximates to a delayed step, but that as the range increases the sharp rise disappears, the time rate of change diminishing rapidly with distance.

The magnitude of the initial step at $\bar{r} = \bar{t}$ (i.e. at $t = \frac{r}{v}$) is

$$e^{-\bar{r}} \quad (1 + \bar{r} + \frac{\bar{r}^2}{2})$$

which is almost zero for $\bar{r} = 10$. In sea water, for which we may take $\sigma = 4 \text{ ohms}^{-1} \text{ m}^{-1}$, $\mu = 4 \times 10^{-7} \text{ Hm}^{-1}$, and $\varepsilon \approx \frac{1}{36} \times 10^{-9} \text{ Fm}^{-1}$, we have $r_o = 1.2$ cm and $t_o = .35$ ns; so the step vanishes at ranges greater than about 12 cm from the source.

We shall be concerned with ranges and times which are several orders greater than r_o and t_o. Under these conditions (1) and (2) reduce to

$$H_\phi \approx \frac{d\ell \sin\theta}{4\pi} \frac{\sqrt{(\mu\sigma)^3}}{4\sqrt{\pi}} \frac{r \exp(-\mu\sigma r^2/4t)}{\sqrt{t^5}} \tag{3}$$

and

$$H_\phi = \frac{d\ell \sin\theta}{4\pi} \{ (\frac{\mu\sigma}{\pi})^{\frac{1}{2}} \frac{\exp(-\mu\sigma r^2/4t)}{r\sqrt{t}} + \frac{1}{r^2}[1-\text{erf}(\mu\sigma r^2/4t)]\} \tag{4}$$

for a δ-function and a step of current respectively. These results could have been derived by putting $\varepsilon = 0$, but they are valid for any ε provided that $t >> \alpha$ and $\beta t >> 1$. In fact, a finite value of ε ensures that the propagation velocity is everywhere finite, whereas (3) and (4) imply an infinite speed near the source.

Ring Source

The field due to an impulsive current in a ring of radius "a" can be deduced from the dipole results given above. If the range considered is much greater than the size of the ring (which will certainly be so in the application we are contemplating) it is found that

$$W_{H_z} = \frac{a^2\sqrt{(\mu\sigma)^3}}{8\sqrt{\pi}} (1 - \frac{\mu\sigma\rho^2}{4t}) \frac{e^{-\xi/t}}{\sqrt{t^5}} \tag{5a}$$

$$W_{H_\rho} = \frac{a^2\sqrt{(\mu\sigma)^5}}{32\sqrt{\pi}} \rho z \frac{e^{-\xi/t}}{\sqrt{t^7}} \tag{5b}$$

$$W_{E_\phi} = \frac{a^2\sqrt{(\mu^5\sigma^3)}}{32\sqrt{\pi}} (5 - \frac{2\xi}{t}) \frac{e^{-\xi/t}}{\sqrt{t^7}} \tag{5c}$$

where the coordinates are now cylindrical (ρ, ϕ, z), and the centre of the current ring is at the origin, the plane of the ring being normal to the z axis.

$\xi = \frac{\mu\sigma}{4}(\rho^2 + z^2)$ is proportional to the square of the distance from the source. The field components have been written as W_{H_z}, etc, because they are in fact weighting functions; and, since the E-M equations are linear, the fields due to any current f(t) which is zero for t<0 can be found immediately as

$$H_z(\rho,z,t) = \int_0^t W_{H_z}(\rho,z,x)\, f(t-x)\, dx \tag{6a}$$

$$H_\rho(\rho,z,t) = \int_0^t W_{H_\rho}(\rho,z,x)\, f(t-x)\, dx \tag{6b}$$

and $$E_\phi(\rho,z,t) = \int_0^t W_{E_\phi}(\rho,z,x)\, f(t-x)\, dx \tag{6c}$$

Equations (5) show that the form of the transient field components as a function of time depends strongly on ρ and z. As these become larger the transient persists for a longer time, as well as diminishing in amplitude. The outward velocity of the transient also falls with range, and is different in different directions. For example, the time rate of change of the axial component H_z (ρ=0) reaches a peak value at z when $t = .047\mu\sigma z^2$, and the velocity of propagation of the peak is $1/.094\mu\sigma z$. For $\mu = 4\pi \times 10^{-7}$ henries m^{-1}, σ = 4 $ohms^{-1}$ m^{-1} (typical of sea-water) and z = 100 m, the peak arrives at 2.3 ms and has a speed of 21 km/s at this point. At 1 km the peak occurs at .23s, with a speed of only 2 km/s. This clearly shows the diffusion-like character of the propagation, which is dominated by the conductivity term.

The transfer functions for the field components - i.e. the Laplace transforms of the weighting functions - can be obtained from (5) as

$$F_{H_z}(s) = \frac{a^2(2z^2-\rho^2)}{4\sqrt{(\rho^2+z^2)^5}}\left[1 + 2\sqrt{(\xi s)} - \frac{4\rho^2\xi s}{2z^2-\rho^2}\right]\exp(-2\sqrt{(\xi s)}) \tag{7a}$$

$$F_{H_\rho}(s) = \frac{3a^2\rho z}{4\sqrt{(\rho^2+z^2)^5}}\left[1 + 2\sqrt{(\xi s)} + \frac{4}{3}\xi s\right]\exp(-2\sqrt{(\xi s)}) \tag{7b}$$

and

$$F_{E_\phi}(s) = \frac{-a^2\mu\rho}{4\sqrt{(\rho^2+z^2)^3}}\, s\left[1 + 2\sqrt{(\xi s)}\right]\exp(-2\sqrt{(\xi s)}) \tag{7c}$$

and putting $s = i\omega$ gives the steady-state results for a sinusoidal source current.

THE USE OF E-M FIELDS IN SHIP POSITION CONTROL

The depths of interest for sea-bed drilling operations are from 100m - 300m. If at these ranges one or more of the field components can be detected for reasonable values of power supplied to the source, we have in principle a direct replacement for the acoustic system. The pulsed E-M method gives arrival time differences of the same order (ms) as for the acoustic case, and could therefore be used in exactly the same way to derive error information. Either a linear element or a solenoid as a detector would give a voltage proportional to the rate of change of the magnetic field; alternatively, a linear array could be employed to detect the electric field.

In addition to this method based on pulse arrival times, there is the possibility of using amplitude and phase characteristics for a sinusoidal excitation of the source. From (7), the amplitudes of H_z and H_ρ for a current ring are

$$|H_z(\omega)| = \frac{a^2(2z^2-\rho^2)}{4\sqrt{(\rho^2+z^2)^5}}\left[1 + 2x + 2x^2 - \frac{4\rho^2}{2z^2-\rho^2}x^3 + \frac{4\rho^4}{(2z^2-\rho^2)^2}x^4\right] e^{-x} \qquad (8a)$$

and

$$|H_\rho(\omega)| = \frac{3a^2\rho z}{4\sqrt{(\rho^2+z^2)^5}}\left[1 + 2x + 2x^2 + \frac{4}{3}x^3 + \frac{4}{9}x^4\right]^{\frac{1}{2}} e^{-x} \qquad (8b)$$

where we have written

$$x = (2\xi\omega)^{\frac{1}{2}} = \left(\frac{\mu\sigma\omega}{2}\right)^{\frac{1}{2}} (\rho^2+z^2)^{\frac{1}{2}}$$

If the transmitting coil or solenoid is at a depth z below the surface with its axis vertical, the magnitude of the voltages induced in vertical and horizontal coils located at any point on a circle of radius ρ centred on a point vertically above the transmitter will be proportional to $\omega|H_z|$ and $\omega|H_\rho|$ respectively. In particular, for $\rho = 0$ we have $|H_\rho| = 0$ and

$$\omega|H_z| = \frac{\omega a^2}{2z^3}(1 + 2x + 2x^2)^{\frac{1}{2}} e^{-x}$$

so for any z there is an optimum frequency which maximises the received voltage, given by

$$\omega = \frac{16}{\mu\sigma z^2}.$$

If account is taken of the "Q" of a tuned receiving coil, the voltage is more nearly proportional to $\omega^2|H_z|$, in which case the optimum frequency is $\omega = 48/\mu\sigma z^2$.

For $z = 100$m, this is 152 Hz, and for $z = 300$m it is 17 Hz. These very low frequencies result of course from the attenuation factor

$$\exp - \left(\frac{\mu\sigma\omega}{2}\right)^{\frac{1}{2}} z.$$

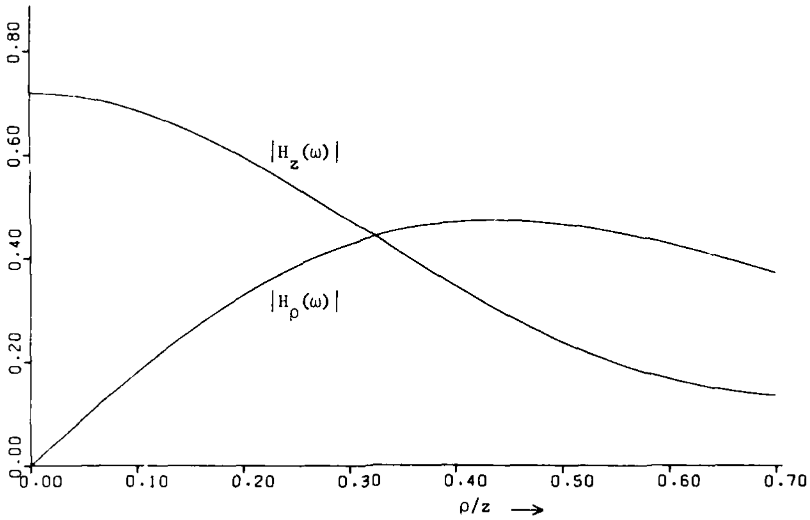


Figure 2. Magnetic Field Amplitudes due to a Current sinωt in a Coil with its Axis in the Vertical (z) Direction, for $(\frac{\mu\sigma\omega}{2})^{\frac{1}{2}} z = 2.5$

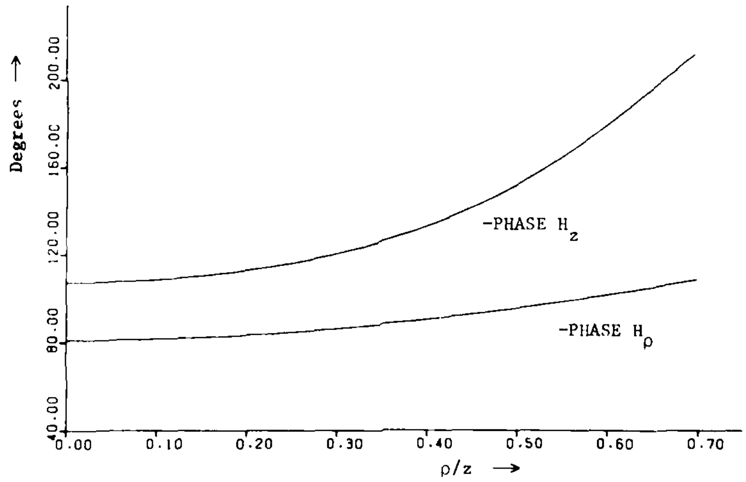


Figure 3. Phase of $H_z(\omega)$ and $H_\rho(\omega)$ for $(\frac{\mu\sigma\omega}{2})^{\frac{1}{2}} z = 2.5$

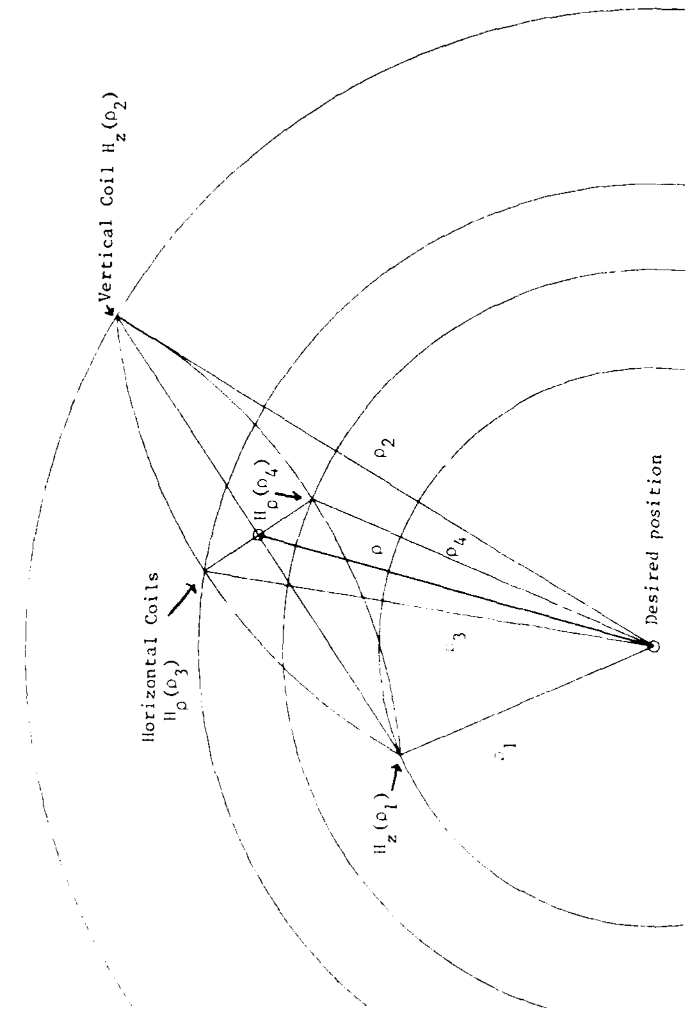


Figure 4. Error Measurement in Ship Axes

Fig. 2 exhibits $|H_z|$ and $|H_\rho|$ as functions of ρ/z. On the sea surface the contours of constant $|H_z|$ and $|H_\rho|$ are circles centred on the point vertically above the transmitter, the former decreasing and the latter increasing with distance. These contours could be utilized as shown in Fig. 4, where the ship is equipped with four receivers: a vertical coil at the bow measuring $|H_z|_1$, say, and a similar coil at the stern giving $|H_z|_2$, with two horizontal mutually perpendicular coils on both the port and starboard sides, measuring $|H_\rho|_3$ and $|H_\rho|_4$ respectively. The fore and aft receivers give voltages proportional to ρ_1 and ρ_2 (Fig. 4) which in turn give the error in ship axes both in magnitude and direction, apart from a 180^o ambiguity. This is resolved by the lateral receivers: the desired position lies to starboard if the starboard receiver gives the smaller reading and vice versa.

The derivation of the error vector in ship axes avoids the need for axis transformation and the measurement of ship's heading with respect to a space datum; the effects of yaw are automatically accounted for in the measurement system. For example, when the ship is in the desired position the measured error remains zero for all yaw angles and no power need be wasted in maintaining an adverse heading with respect to wind, wave or tide.

We have so far taken no account of changing z (due to surge) or of pitch and roll. However, there is sufficient redundancy in the system outlined above to eliminate their effects - in fact, the received signals furnish an estimate of z as well as the error ρ. There is also further information available in the phase of the signals: these are shown in Figure 3.

SEA SURFACE EFFECTS

The foregoing results are derived on the assumption that the conducting medium is infinite in extent, so both source and field points are taken to be remote from boundaries with different media. This however is by no means the case for the application which we have in mind; the transmitter will be located near the sea floor, while the receiver will be just below, or above, the sea surface.

The presence of such boundaries leads to considerable complication, since the field in each medium depends on the properties of both media. Moreover, there are four different cases to consider: magnetic or electric dipole transmitters, with the dipole axis either normal to or parallel with the surface separating the two media (the field from any other source can be obtained as a linear combination of the fields due to these four sources).

For the general case in which both media have different conductivities, emissivities and permeabilities, the cylindrical components of the fields can be formulated as integrals involving Bessel functions. If the two media are salt water and air the conductivity of the former is the dominant factor, and the emissivities of both media can be neglected. In this situation analytical solutions have been found for the surface field components due to any of the four sources mentioned above when they are excited harmonically. Comparison with computer solutions of the full equations have confirmed that the approximate solutions are very accurate at the low frequencies of interest.

In general the total field may be ascribed to the direct field from the source (as if there were no surface), a similar field from an image source, and a third source due to electrical currents distributed over the surface. Since we have assumed the same magnetic permeability for each medium, all components except E_z are continuous at the surface. To illustrate the effect of the sea-air surface we take the particular case of a vertical magnetic dipole as the source,

ON IDENTIFICATION OF NONLINEAR SPEED EQUATION FROM FULL SCALE TRIALS

by
M. Blanke

Technical University of Denmark
Lyngby, Denmark

ABSTRACT

The paper describes a novel approach on identification of the nonlinear dynamic speed equation for a surface ship from full scale measurements. Different methods of analysis are discussed, and the application of frequency analysis techniques is shown to be attractive regarding separation of the effects of different nonlinear terms and concerning measurement of small amplitude loss terms.

The findings reported are the relative large influence on propulsion from steering activity due to wake fluctuations and the practical demonstration of hydrodynamic memory effects being important.

The dynamical speed equation is finally discussed in the paper concerning autopilot performance.

INTRODUCTION

Speed loss of vessels due to steering activity is achieving an increasing interest because significant reductions in operating costs can be obtained by even minor improvements in propulsion efficiency during the long periods of automatic course keeping at sea.

Loss of speed is due to the movements of the hull and deflections of the rudder but also the influence on propeller performance from sway and yaw is contributing. In undisturbed conditions, the average speed loss due to rudder activity and hull motions can accurately be expressed using existing mathematical models ([1] and [2]), but under dynamic conditions the effects of varying inflow velocity to the propeller and its interaction with the engine must be considered. Such dynamic conditions are met e.g. when waves and wind influence the coursekeeping.

The accurate modelling of the combined dynamic performance of ship, propeller and engine is essential when optimum performing autopilots are to be designed, in particular when the governor control loop is to be attached to the course keeping loop as to achieve minimum propulsion losses.

The paper is concerning verification of such a mathematical model by means of steering experiments on full scale ships.

medium is to introduce a small phase change in H_ρ, gradually increasing with increasing ρ, and to smooth out the sudden phase reversal of H_z, the degree of smoothing depending on the value of σ.

For these experiments the measured conductivity was 2.5 reciprocal ohms per metre, and Figures 12(b) - 15(b) show the theoretical curves of phase as a function of ρ for this value of σ, together with the experimentally measured values at 3 ft intervals. The transmitter and receiver angles are those given above for the amplitude plots.

It will be seen that when the receiver is fully submerged the measured and theoretical values for the phase of H_z are in close agreement (Fig. 13(b)). As the distance from the source increases (Figs. 12(b) and 14(b)) the agreement becomes less good: however, the amplitudes for large ρ are then very small, so it is difficult to obtain a good measure of phase (the ratio of two small quantities) with the analogue equipment used.

The expected change in the phase of H_ρ (Fig. 15(b)) is less than 1 degree, smaller than the resolution of the signal processing unit. Nevertheless, the increasing trend in the experimental points is clearly matched by the theoretical curve.

CONCLUDING REMARKS

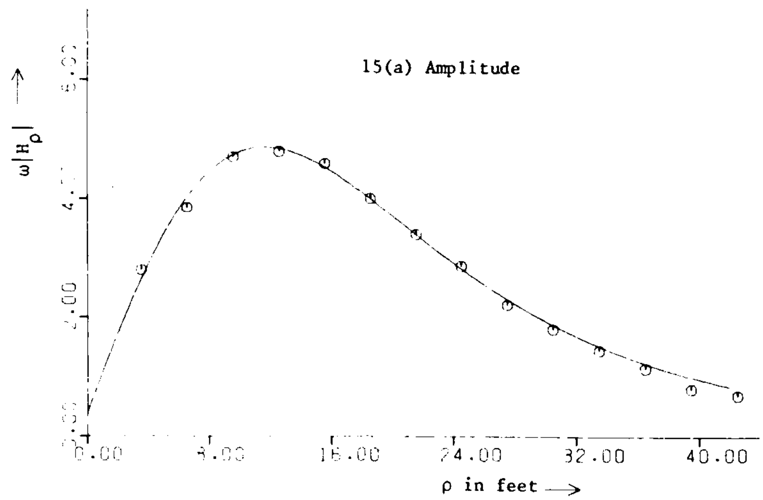
The investigations outlined in this paper suggest that underwater electromagnetic transmission provides a feasible means of error detection for a ship position control system, and might well offer advantages in terms of accuracy, simplicity and reliability. The transmitter depth required for an operational system is of course greatly in excess of that available for the experimental work described in the paper, although the frequency used (23 Hz) is appropriate for depths of 100 - 300 m. Further work is planned to gather data under conditions closer to those in which the apparatus would actually be used as part of a control system; in particular, the measurement of the signal to noise ratio for greater depths and for various frequencies.

It is also proposed to examine other components of the electromagnetic field for both magnetic and electric dipole sources. Because of the complexity of the ship control problem it may be desirable to take advantage of the additional information available (electric and magnetic components, horizontal and vertical, and their phases relative to each other and to the source) by detecting, correlating and filtering the signals in an optimum way, taking account of the different noise structure of each component.

ACKNOWLEDGEMENTS

The authors are greatly indebted to Dr. P.F. Blackman, who designed the transmitter and receiver used in the experiments; and to Mr. E.H. Bignall, who made all the equipment involved.

They are also very grateful to Mr. F. Hodgson, Mr. E. Davis and Mr. D. Doe, who provided the facilities at Horsea Lake and gave their generous assistance during the trials.



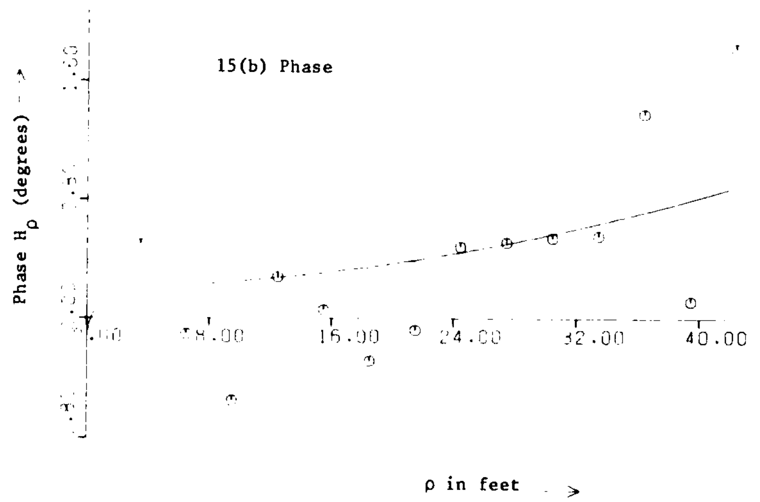


Figure 15. Receiver near-horizontal at water surface: Corrected Results
Depth of transmitter 24.13 ft.
Transmitter axis 0.045 radians from vertical
Receiver axis 0.015 radians from horizontal

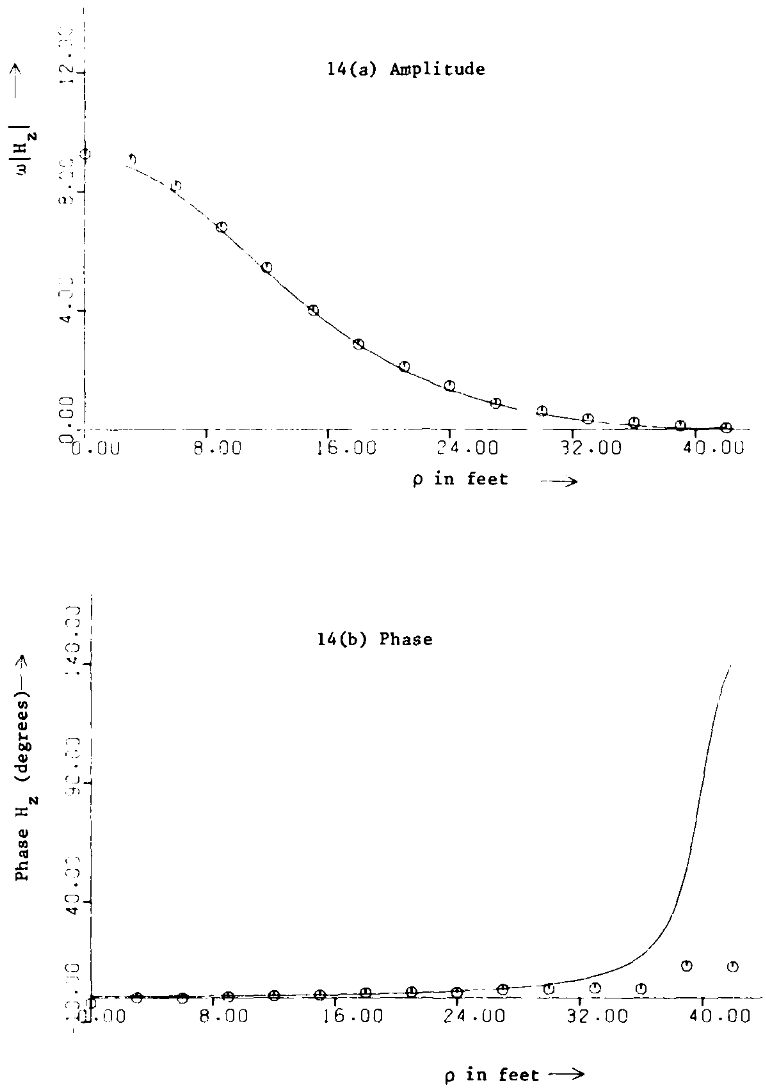


Figure 14. Receiver fully out of water: Corrected Results
Depth of transmitter 24.13 ft
Transmitter axis 0.1 radians from vertical

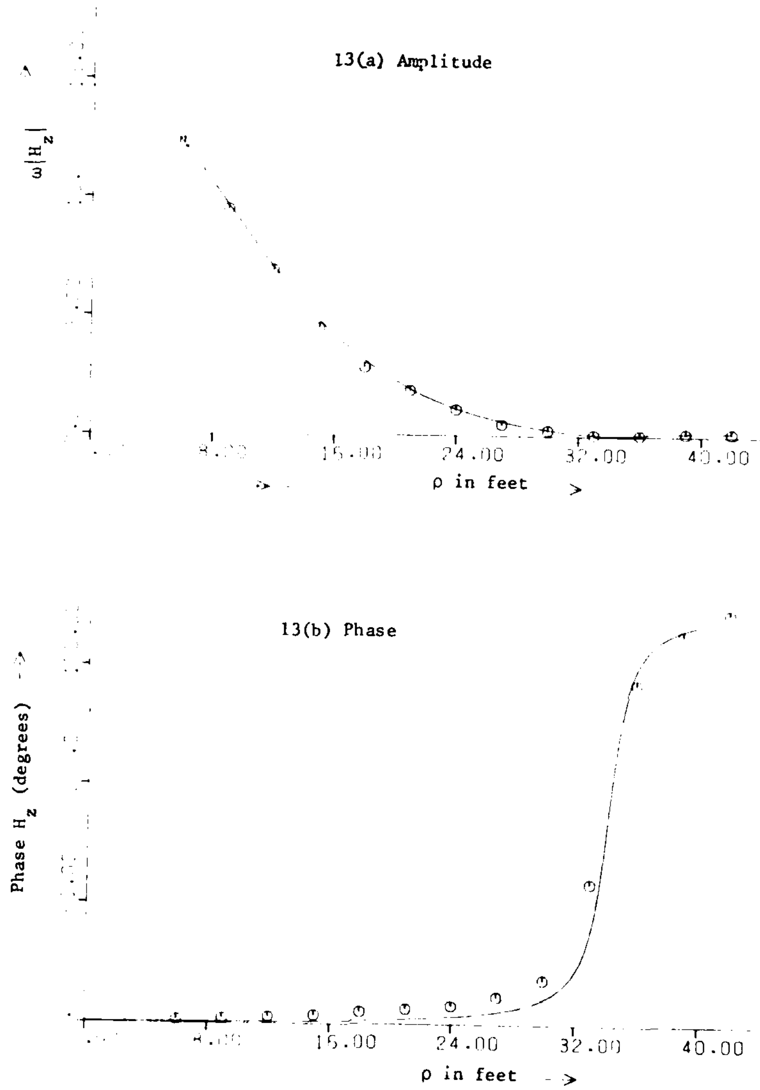
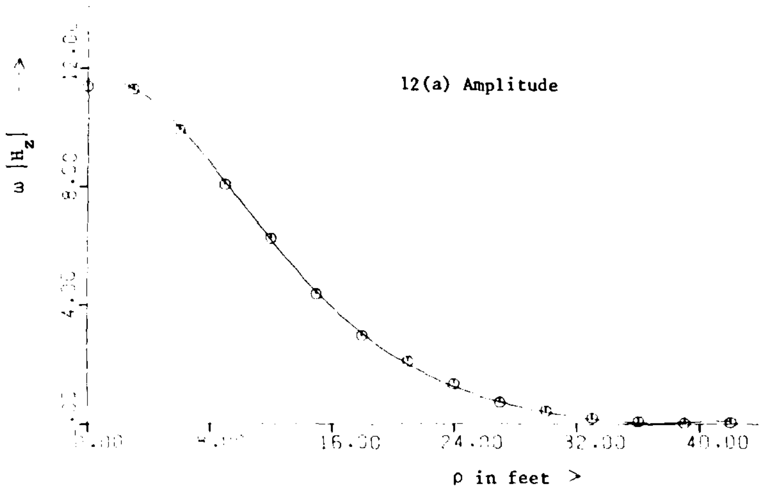


Figure 13. Receiver just submerged: Corrected Results
Depth of transmitter 24.13 ft.
Transmitter axis 0.1 radians from vertical



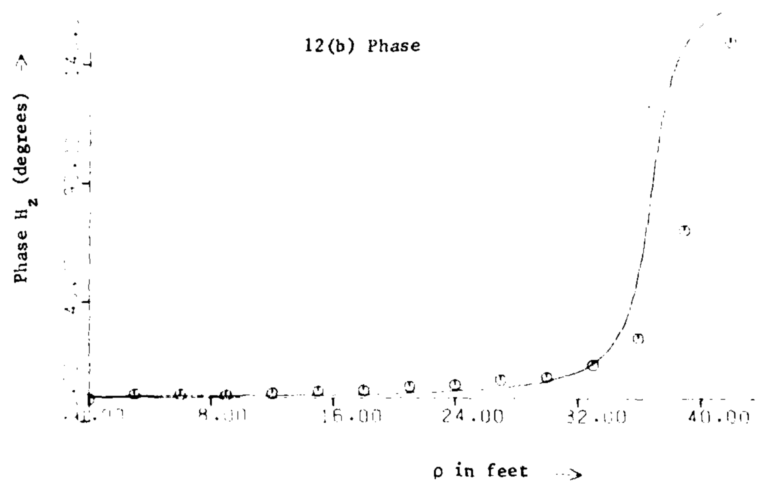


Figure 12. Receiver Halfway in water: Corrected Results
Depth of transmitter 24.13 ft
Transmitter axis 0.1 radians from vertical

The transmitter was suspended vertically by its cable from a small dinghy, the lower end of the transmitter being attached to a weight resting on the bed of the lake in order to minimise transmitter motion due to currents. To the same end an attempt was made (not wholly successfully) to prevent dinghy motion by means of lines from dinghy to shore. The receiver was on a raft which could be moved in a fixed direction and at known distances from the point of suspension of the transmitter. To avoid saturating the receiver at the short distances involved (maximum 42 ft) it was necessary to reduce the transmitter supply voltage from 150V to 50V, so the power consumption of the transmitter was rather less than 1 watt.

Readings of H_z and H_ρ (or rather their time rates of change) were taken at intervals of 3 ft, measured along the surface from the point vertically above the transmitter. At each such value of ρ the vertical component H_z was measured at three points: with the receiver (suspended from its cable) just submerged, halfway out of the water and fully out of the water. The horizontal measurement H_ρ was taken with the receiver just below the surface, and held horizontal (as near as could be judged) by means of ropes lashed to receiver and raft.

Amplitude Measurements

The results so obtained are shown in Figures 8 - 11, where the full lines are the theoretical curves. The differences in the three sets of readings for the amplitude of H_z (Figures 8, 9 and 10) are not of course due to surface effects: they are merely the result of changes in the z coordinate.

While agreement between theory and experiment is fair, the results are less accurate than expected, even allowing for the somewhat rudimentary nature of the experimental arrangement. Later investigation however showed that the distance measurements were in error, because the datum points used on transmitter and receiver housings were displaced from the centres of the coils. These errors (4" for the transmitter and 6" for the receiver) would of course be quite insignificant at ranges of 100 metres or so, but they become important at the short distances involved in these experiments.

Figures 12 - 15 show the effect of correcting the distance measurements, and also of introducing small angles to account for the possibility that the transmitter axis was not exactly vertical and that the receiver axis was not exactly horizontal for the H_ρ measurements. The best fit to the H_z data was found to be with a deviation of the transmitter from the vertical of 0.1 radians. For the H_ρ data, which was taken the following day, the theoretical curves shown are for a transmitter angle of 0.045 radians from the vertical and a deviation of the receiver axis of 0.015 radians from the horizontal.

Angular errors of this magnitude could certainly have existed. With the above adjustments the differences between the experimental points and the theoretical values (the full lines in Figures 12(a) - 15(a)) are very small, and are within the limits to be expected from the known inaccuracies inherent in the analogue equipment (multipliers, operational amplifiers etc.) used in the signal processing unit.

Phase Measurements

As mentioned earlier, the conditions for the experiments described above are such that the amplitude of H_z and H_ρ are almost exactly as they would be in air. This however is not true of the phases: these begin to show differences before any distinction between amplitudes becomes apparent. In air, at the very long wavelengths involved, the phase of H_ρ with respect to the transmitter does not change with ρ, while H_z simply reverses sign at $\rho = \sqrt{2}z$. The effect of the conducting

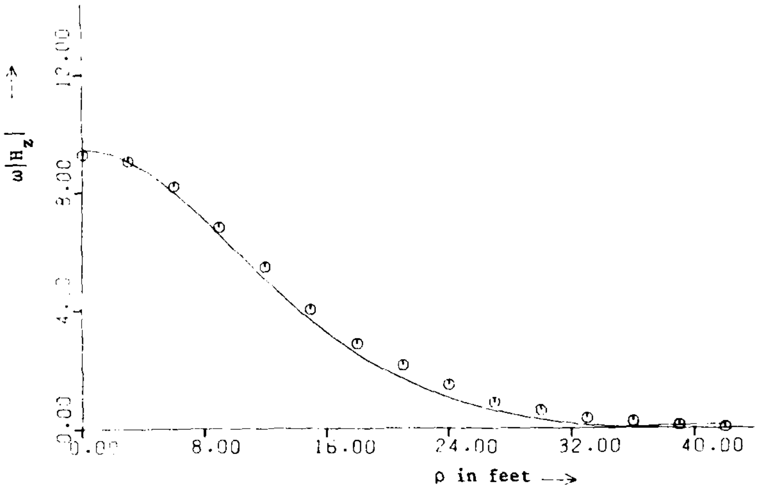


Figure 10. Receiver fully out of water
Depth of transmitter 23.75 ft.
Transmitter axis assumed vertical

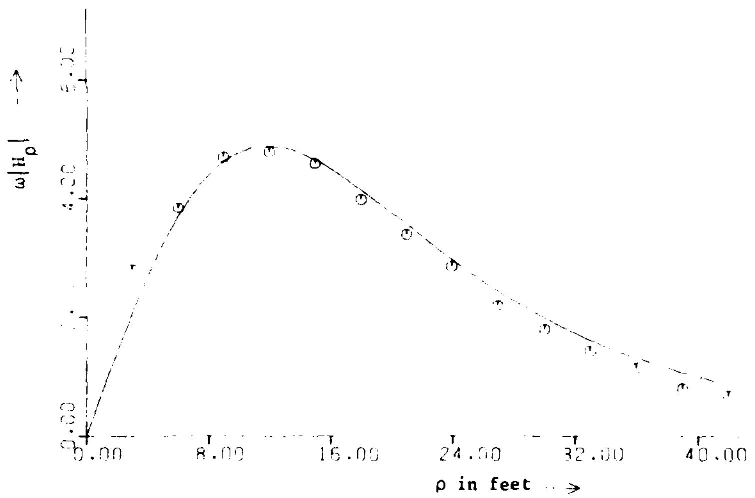


Figure 11. Receiver horizontal near water surface
Depth of transmitter 23.75 ft
Transmitter axis assumed vertical
Receiver axis assumed horizontal

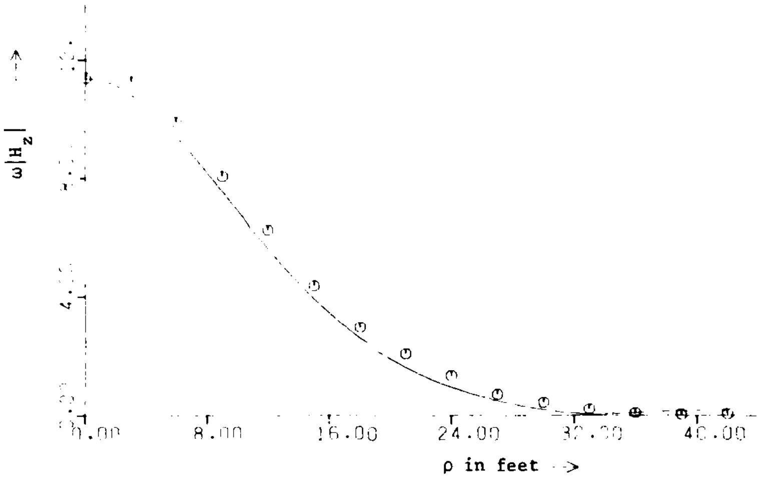


Figure 8. Receiver Halfway in Water
Depth of transmitter 23.75 ft
Transmitter axis assumed vertical

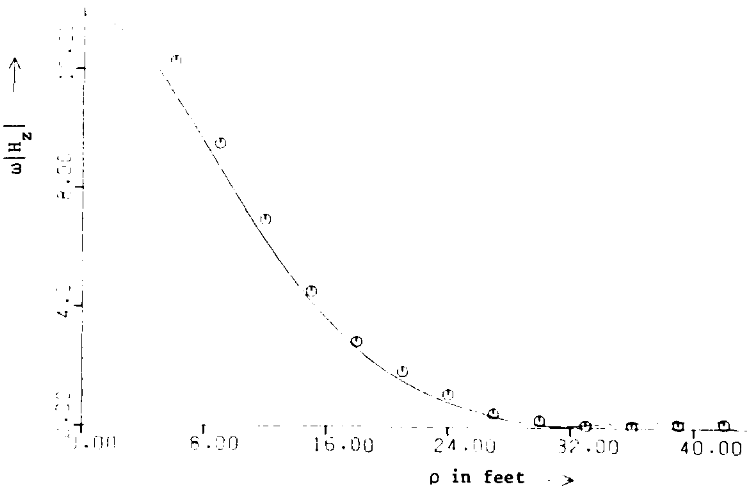


Figure 9. Receiver just submerged
Depth of transmitter 23.75 ft.
Transmitter axis assumed vertical

mitter, but the coil has 20,000 turns, giving an inductance of 136 henries and therefore a tuning capacity of about .35μF. The receiver unit also contains a remote tuning device and an active filter network, thus ensuring that the signal transmitted through the cable to the signal-processing unit is at a reasonable voltage level.

The tuning is controlled from the far end of the cable by a DC voltage which alters the effective resistance of a field-effect transistor incorporated in the feedback circuit of an operational amplifier. There is also positive capacitative feedback, so the effective capacitance across the receiver coil terminals is adjustable via the DC voltage. This arrangement has the advantage that the cable wiring is not included in the tuned circuit, thus obviating a possible source of noise.

Further operational amplifiers within the receiver tube provide a filter with the transfer function:

$$\left(\frac{k_1 sT}{1 + k_2 sT + s^2T^2}\right)^2$$

where k_1, k_2 are chosen to give a high gain at $\omega = 2\pi.23 = \frac{1}{T}$ and a band width of a few Hertz about this frequency.

Signal Processing Unit

As mentioned above, the transmitter power is supplied remotely from a DC source. The firing of the transistor gives rise to a small current pulse down the cable, and from this (after filtering) a sine wave can be derived, its phase being linked to that of the transmitter coil current, and thus to the phase of the generated magnetic field.

If $A\sin\omega t$ is the reference signal so obtained, and the received signal is $V\sin(\omega t+\phi) + N(t)$, where N is noise, cross-correlation of the two gives

$$\overline{(V\sin(\omega t+\phi) + N(t))A\sin\omega t} = AV\cos\phi$$

provided signal and noise are uncorrelated.

The multiplication is effected in the signal processing unit by an analogue multiplier, and the averaging by a simple CR network with an adjustable time constant (2-10s). A second multiplier is arranged to give $V \sin\phi$ (by using the reference signal shifted by 90°), so smoothed values of V and ϕ can be obtained as DC voltages.

EXPERIMENTAL RESULTS

In order to test the validity of the theoretical work, the apparatus described above has been used in a sheltered salt-water lake (Horsea Lake, Portsmouth), and the results so obtained are presented in Figures 8 - 15.

The experimental equipment was designed to operate at depths of up to 100 metres: unfortunately the greatest depth available at the salt-water lake used for the trials was only about ten metres. For h = 10m and f = 23Hz, the parameter $\bar{h} = 0.15$, for which the amplitudes of H_ρ and H_z are almost exactly as they would be in air, the presence of the water and of the surface having a negligible effect (Figure 5a). (There is however a detectable difference in phase: this is discussed below).

and consider the amplitudes of the two components (H_z and H_ρ) of the magnetic field at the surface. The fields can be conveniently expressed as functions of two non-dimensional parameters $\bar{h} = kh$ and $\bar{\rho} = k\rho$, where h is the depth of the source below the surface and $k = (\frac{\mu\sigma\omega}{2})^{\frac{1}{2}}$. The factor $K = 4\pi/Mk^3$ where M is the magnetic moment of the source.

Figures 5, 6 and 7 show the amplitudes of H_ρ and H_z at the surface for $\bar{h} = 0.2$, 0.8 and 2.0 respectively; in each case curves are also given for the response in air (i.e., ignoring the conductivity term) and for that in an infinite medium (ignoring the presence of the surface). It will be observed that, for $\bar{h} = 0.2$, the three curves are identical, so the field is virtually independent of the medium. For $\bar{h} = 0.8$ the curves begin to diverge, while at $\bar{h} = 2.0$ they are distinctly different. It is interesting to note that, for $\bar{h} = 2.0$, the presence of the surface reduces the value of H_ρ for $\bar{\rho}$ less than about 3.5, but for H_z the reverse is true - the surface enhances the amplitude of this component.

If we take $\sigma = 4\,ohm^{-1}m^{-1}$ and f = 25Hz, then k is about 0.02, and the curves are for h = 10, 40 and 100 metres; so with these parameters the surface effects become noticeable at source depths greater than about 40 metres. At a given source depth the effects become more apparent as the frequency (or the conductivity) is increased - e.g., at f = 100 Hz the figures represent the situation at depths of 5, 20 and 50 metres.

EXPERIMENTAL APPARATUS

As mentioned earlier, the optimum frequency for the depths of interest for ship control (100 - 300m) ranges from about 150Hz down to 15Hz. In order to investigate the feasibility of the method an experimental system has been constructed to operate on a frequency of 23Hz: this should give detectable signals at ranges of 100m or more, depending on the ambient noise level.

Both transmitter and receiver units are enclosed in sealed plastic tubes, to which are attached cables for the supply of power to the transmitter and the extraction of signals from the receiver. These cables (100m for the transmitter and 25m for the receiver) also act as supports, and are arranged in such a way that there is no strain on the cables at the points where they enter the tubes. The complete unit can operate on dry batteries where mains power is not available.

Transmitter Unit

The transmitter coil consists of 635 turns of wire wound on a laminated iron core having a circular cross-section 3 cm in diameter and a length of 91 cm. The coil inductance is 0.4 henries; a capacity of 120µf is therefore needed to give a natural frequency of 23Hz. The twelve 10µF capacitors are housed alongside the coil in the plastic tube, which also contains a drive circuit to maintain oscillations at a constant level. The circuit is fed via the 100m cable from a 150 V DC supply, and a current of 50 mA (i.e. a power consumption of 7.5W) is sufficient to give a B_{max} of about 10 kilogauss (near saturation of the iron). The external magnetic field due to the transmitter is proportional to the dipole moment Bv, where v is the volume of the iron; so the field can be increased by increasing v, and at the same time providing more power to accommodate the increased iron loss (also proportional to v for a given B). The dimensions chosen here are a compromise between the desire for a high field strength and the need to minimise the weight for reasonable portability. (The transmitter unit weighs 17 kg).

Receiver Unit

The receiver, weighing 11 kg, has an iron core identical to that of the trans-

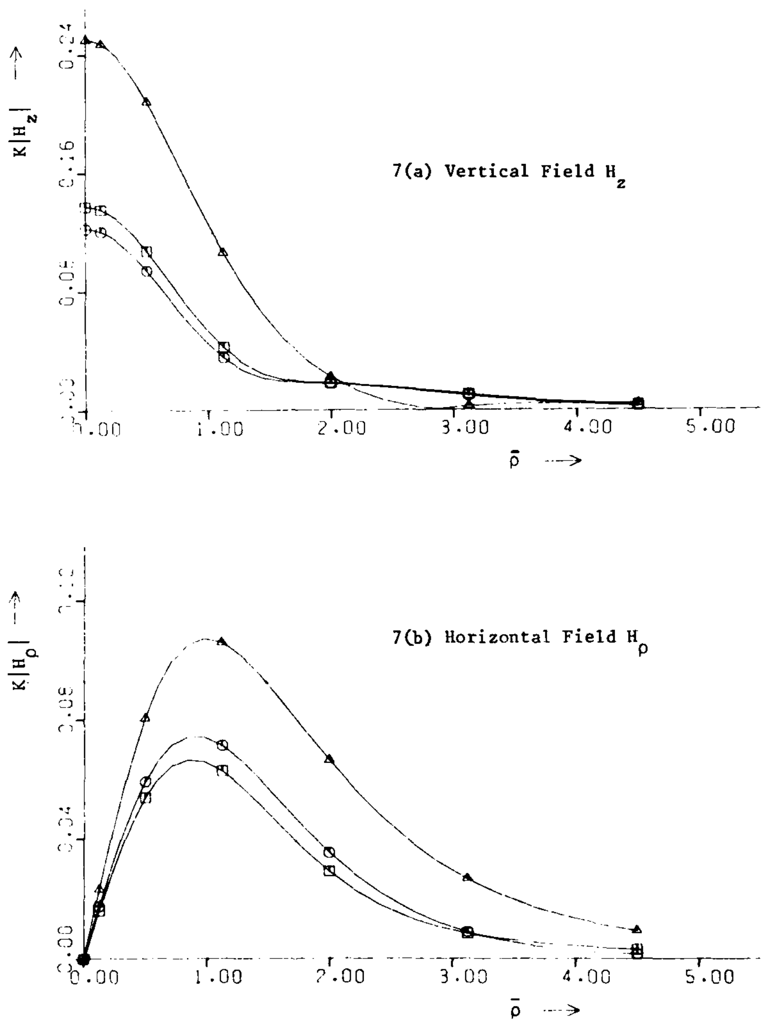
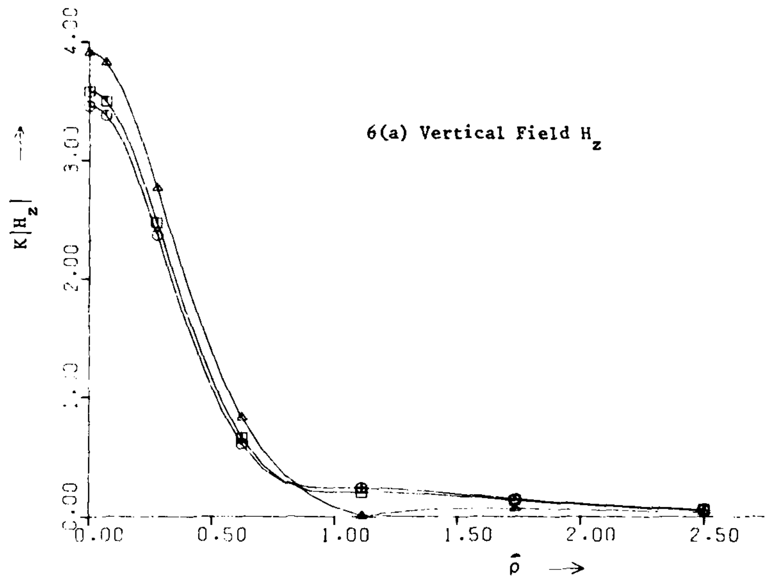


Figure 7. Amplitude of Magnetic Field for $\bar{h}$ = 2.0

- △ In air
- ○ In an infinite conducting medium
- □ In a conducting half-plane with an air interface



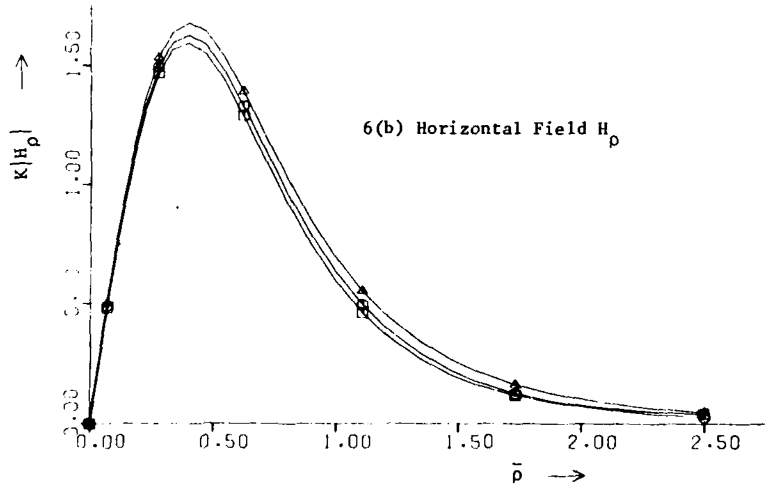


Figure 6. Amplitude of Magnetic Field for $\bar{h} = 0.8$

▲ In air
⊙ In an infinite conducting medium
⊡ In a conducting half-plane with an air interface

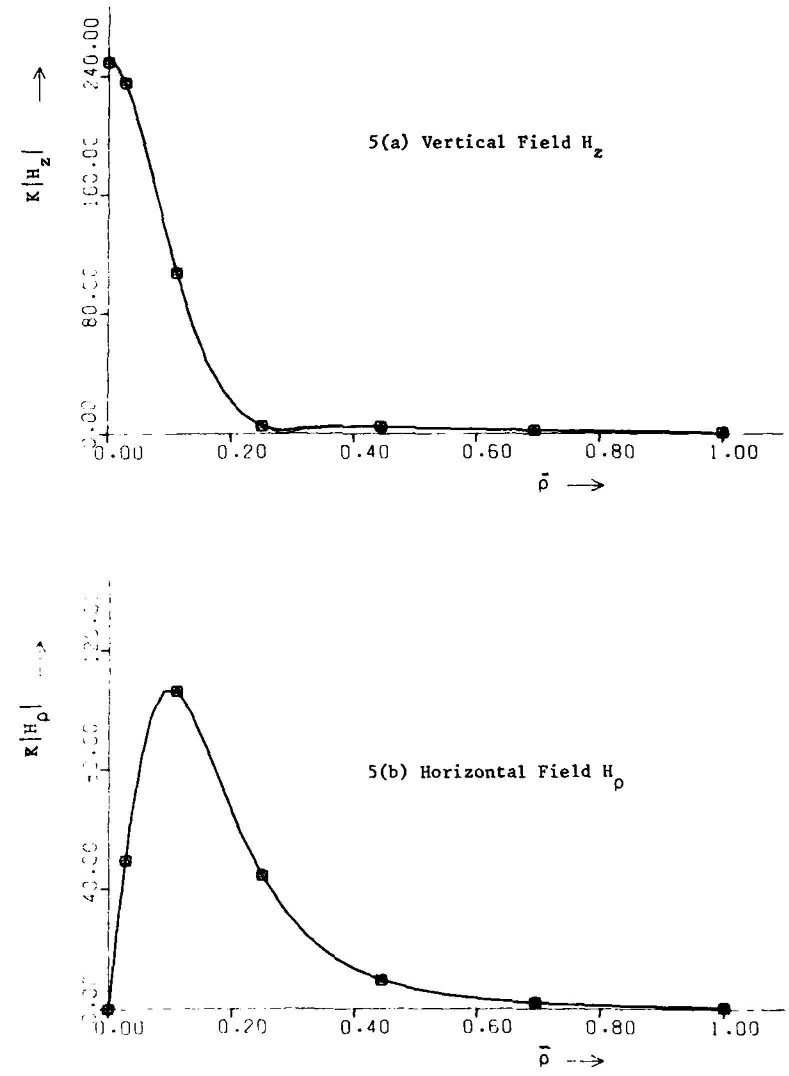


Figure 5. Amplitude of Magnetic Field for $\bar{h} = 0.2$

- △ In air
- ○ In an infinite conducting medium
- ☐ In a conducting half-plane with an air interface

A MATHEMATICAL MODEL

The dynamics of ship manoeuvring has been treated in various papers. Mostly, the mathematical models are developed for predicting ship steering qualities and for the programming of ship handling simulators. Those models are able to be adapted to a variety of ship types and applicable over a wide range of ship speeds and engine revolutions. They give precise representations of manoeuvring responses provided a proper set of coefficients is obtainable. The equations appear as quite complex as numerous derivatives, coefficients and characteristics are employed which require sophisticated techniques for being determined.

When autopilot design is considered, a dynamical model must be available, which exactly describes the structure of the system to be controlled.Small uncertainties in model parameters generally do not influence closed loop performance, whereas lack of information on couplings between parts of the system may prevent the proper function of a controller. A precise model having too many parameters which are hardly obtainable is therefore less feasible than one which gives approximate responses only but has the correct structure and is described by obtainable parameters. Such simplified mathematical models are derived from the more advanced descriptions by e.g. considering only a limited range of operational conditions. For course keeping autopilot design, the simplified yaw equation is an example of such a model which has been used during more than a decade.

Models of reduced complexity are also required when free sailing full scale experiments are used for model validation, because the parameters in the complete descriptions may not be able to be identified separately. Instead, the composite coefficients entering in descriptions on state variable form or in the transfer functions can be determined. Which parts of the equations may actually be identified depend strictly on the actuators and measurements available for the experiments.

Most control systems design procedures are based on models being on state variable form or given as transfer functions from inputs to outputs. When a model is derived on this form, it is thus feasible for being used both as a basis of autopilot design and for model validation by input-output experiments.

Aiming design of autopilots which minimize propulsion losses a model of the ship steering dynamics which include interaction with screw and engine is outlined below. The model is derived from more complete descriptions by assuming a constrained set of motions of the hull in order to relax the equations from certain extraneous nonlinear terms. The form of the model is also chosen to be adequate for identification of the propulsion dynamics by free sailing experiments.

Ship Dynamics

For completely describing the motions of a ship six degrees of freedom are required, translations in three directions and rotations around three axes. When employed for describing the propulsion loss mechanisms, however, it is sufficient to consider sway, yaw and surge motions only, as the pitch, heave and roll dynamics form a set of equations which are only insignificantly coupling to the remainder of the motional equations.

In closed loop autopilot operation, coupling between roll and yaw can be induced through rudder activity in certain conditions. For full form merchant ships the coupling is rather small and is mainly perceptible around the natural roll eigenfrequency. Generally this is much higher than the band of frequencies where propulsive loss effects are preponderant, so by proper autopilot design the effect can be suppressed. The roll equation is therefore further on not considered.

The formulation of the equations is well known from the thorough description by Abkowitz [1] and Norrbin [2]. It is not intended to repeat details of those works here, but it is feasible to use fragments of their developments as a base for discussing the simplifications possible when the restricted model is derived.

To describe the equations of motion the coordinate system fixed to the ship is shown in fig. 1.

The foundation of the dynamical behaviour of a ship is the conservation of linear and angular momentum. Mathematically this is expressed by the laws of Newton in eq. 1 where only sway, surge and yaw motions are considered.

$$m \cdot \begin{bmatrix} a_x \\ a_y \end{bmatrix}_{abs} = \begin{bmatrix} X \\ Y \end{bmatrix}_{hydrodyn.} + \begin{bmatrix} X \\ Y \end{bmatrix}_{prop} + \begin{bmatrix} X \\ Y \end{bmatrix}_{rudder} + \begin{bmatrix} X \\ Y \end{bmatrix}_{external} \qquad (1)$$

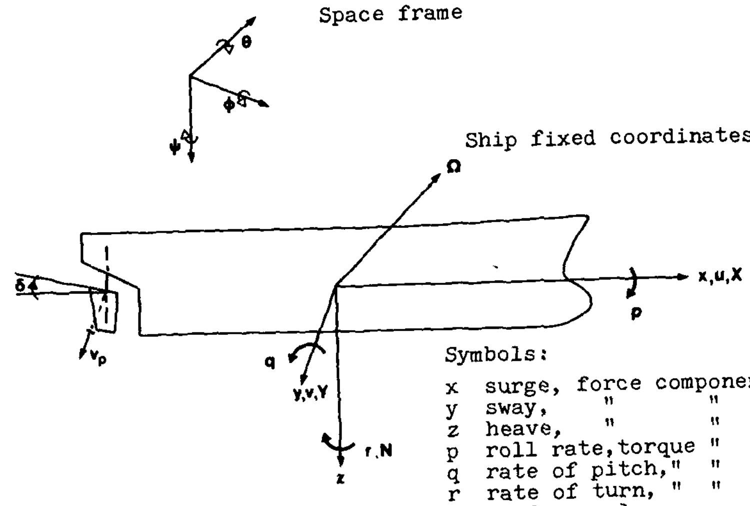
$$I_z \cdot [\dot{\Omega}]_{abs} = [N]_{hydrodyn.} + [N]_{prop} + [N]_{rudder} + [N]_{external}$$

The right hand side force and torque components X, Y and N are separated according to their origin: Hydrodynamical effects, propeller excitation,rudder activity and external disturbances from waves and wind. Ship mass is m and inertia around the vertical axis I_z. Force and torque components refer to the directions of ship fixed coordinates whereas the accelerations $(a_x, a_y, \dot{\Omega})_{abs}$ are relative to an inertial frame.

The accelerations are more conveniently expressed in terms of the velocities relative to ship axes. If x_G is the x coordinate of the center of mass, the momentum vector becomes

Space frame

Ship fixed coordinates

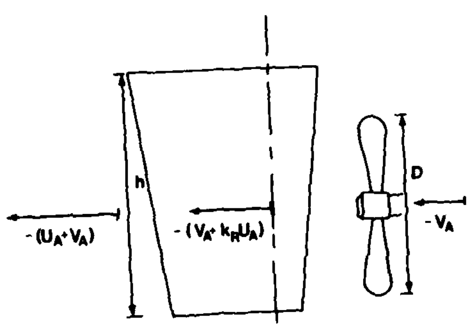


Symbols:

x	surge, force component is		X
y	sway,	"	" Y
z	heave,	"	" Z
p	roll rate, torque	"	" K
q	rate of pitch,	"	" M
r	rate of turn,	"	" N

δ rudder angle
v_p sway velocity at stern
u surge velocity
v sway velocity
Ω angular velocity vector in inertial space.

Fig 1. Definition of ship fixed coordinates and of componentes of force and torque.



Symbols:

D screw diameter
h height of rudder
α is D/h
V_A inflow velocity at screw
U_A flow velocity downstream due to screw.
$V_A+k_R U_A$ flow velocity past rudder

Fig 2. Geometry of screw and rudder. Flow velocities are relative to ship.

$$\begin{bmatrix} ma_x \\ ma_y \\ I_Z\dot{\Omega} \end{bmatrix}_{abs} = \begin{bmatrix} m(\dot{u}-rv-x_G r^2) \\ m(\dot{v}+ru+x_G\dot{r}) \\ I_Z\dot{r}+mx_G(\dot{v}+ru) \end{bmatrix} \tag{2}$$

Below the force and torque components in the motional equation are treated separately according to their origin.

Hydrodynamic Forces. The accelerations are important in the determination of transient hydrodynamic performance of the hull. Because exchange of energy between the hull and the streaming water is involved in changes of ship velocity vector ($\dot{x}$, $\dot{y}$, $\dot{r}$) effects of virtual added mass and added inertia occur in the equations. Terms equivalent to the acceleration components, eq. 2 are in fact present, but the magnitude of each coefficient has to be separately determined.

Steady state velocity also gives rise to resistance in a viscous fluid. The reaction on the hull from the steady forward advance is in essence the resistance of the wet surface. This is proportional to the square of the forward speed. Wavemaking and other effects further increase the resistance at high speeds.

Reactions from steady sway and yaw velocity can be approximated by the readily calculated forces and torques on a slender wing of equivalent dimensions as the hull. Those reactions appear to vary bilinearly with speed times sway or rate of turn. By large excursions in sway and yaw rate, however, nonlinear terms are predominant due to cross flow effects and at high frequencies of excitation, the reactions depend also on the timehistory of motion.

Here the objective is to utilize the equations for operative conditions as course keeping only. The inclusion of the simple bilinear terms therefore suffices for the sway and yaw rate equation.

The hydrodynamic forces employed in the equation of motion thus are

$$\begin{bmatrix} X \\ Y \\ N \end{bmatrix}_{hydrodyn.} = \begin{bmatrix} X_{uu}\,u^2 + X_{vr}\,vr + X_{rr}\,r^2 \\ Y_{\dot{v}}\,\dot{v} + Y_{\dot{r}}\,\dot{r} + Y_{ur}\,ur + Y_{uv}\,uv \\ N_{\dot{v}}\,\dot{v} + N_{\dot{r}}\,\dot{r} + N_{ur}\,ur + N_{uv}\,uv \end{bmatrix} \tag{3}$$

In the equation the derivative $\partial N/\partial u \partial r$ has been abbreviated as N_{ur} etc.

Screw Behaviour. The Propeller action in a field of steady inflow velocity is described by relations between developed thrust T, propeller shaft torque Q_1 and velocity components according to fig.2. By using momentum theory and considering the lift developed by the screw blades, thrust and torque can be shown to depend on bilinear and quadratic terms in screw angular velocity n and inflow velocity at the propeller V_A. During the steady course keeping, fluctuations in propeller load are not excessive compared to the average load. From practical experience then propeller thrust and torque can be readily assessed.

Ship screw performance is commonly presented in nondimension form by the torque and thrust coefficient K_Q and K_T varying over the range of propeller loads as reflected in the advance number J. For the small load variations considered, K_Q and K_T can be expressed satisfactory accurate merely using a linear approximation

$$\begin{aligned} K_T &= K_{TO} + K_{TJ}\, J \\ K_Q &= K_{QO} + K_{QJ}\, J \end{aligned} \qquad (4)$$

where

$K_T = T \,/\, (\rho 4\pi^2 n^2 D^4)$

$K_Q = Q_1 / \,(\rho 4\pi^2 n^2 D^5)$

$J = V_A / \,(2\pi n D)$

D is propeller diameter

n is propeller angular velocity in rad/sec.

In dimensional form we get

$$\begin{aligned} T &= T_{nn}\, n^2 + T_{nv}\, nV_A \\ Q_1 &= Q_{nn}\, n^2 + Q_{nv}\, nV_A \end{aligned} \qquad (5)$$

The inflow velocity at the propeller V_A is related to the ship forward speed by the wake coefficient w as

$$V_A = (1-w)\ u \qquad (6)$$

where u is the ship forward speed.

The propeller action is less predictable when the inflow velocity is varying. Very few surveys have ever been made of the screw behaviour when subject to varying sway velocity, and due to the highly complex boundary layer interaction with propeller and stern, no simple analytical expression describes the phenomena concisely. Empiri-

cally, the effect of local sway velocity at the propeller can be expressed by its influence on the average wake coefficient

$$w = w_o + W(r,v) \tag{7}$$

For a single screw vessel, typically a plot of W(r,v) like fig. 3 will result [7]. The wake is seen to vary as an odd function of local sway velocity in the range normally exploited at steady course. The odd dependancy of wake coefficient from local sway is understood to be the build up of antisymmetric surge and pressure fields due to the one way screw rotation.

The sway at the propeller is a composite of ship sway v and yaw rate r, $v_P = v - l_{PG}r$. The coefficent l_{PG} is the geometric distance from centre of gravity to the propeller position.

Linearization of the wake function is obtained by expanding in first order components in ship sway and rate of turn

$$w = w_o + w_v v - w_v l_{PG} r \tag{8}$$

The inflow velocity to the propeller thus becomes

$$V_A = (1-w_o)\,u - w_v(v-l_{PG}r)\,u \tag{9}$$

Equations (5) and (9) then constitute the description of propeller performance.

<u>Rudder forces.</u> A rudder may be considered a small aspect ratio wing. The lift force then vary with the square of advance velocity at the rudder and for small rudder angles it is proportional to rudder deflection δ. Accordingly rudder drag is proportional to the square of rudder deflection.

Transformation of rudder force to ship fixed coordinates is required for fitting in the equations of motion for the ship. This transformation and approximation of sine and cosine terms by assuming small rudder deflections result in the rudder forces

$$\begin{aligned} X_R &= X_{cc\delta\delta}\,\bar{c}^2\,\delta^2 \\ Y_R &= Y_{cc\delta}\,\bar{c}^2\,\delta \end{aligned} \tag{10}$$

The torque exposed to the hull from the rudder is expressed through the distance l_{RG} from the rudder to centre of gravity

$$N_R = l_{RG}Y_R = l_{RG}\,Y_{cc\delta}\,\bar{c}^2\delta \tag{11}$$

Here $\bar{c}$ is the mean flow velocity at the rudder. The velocity $\bar{c}$

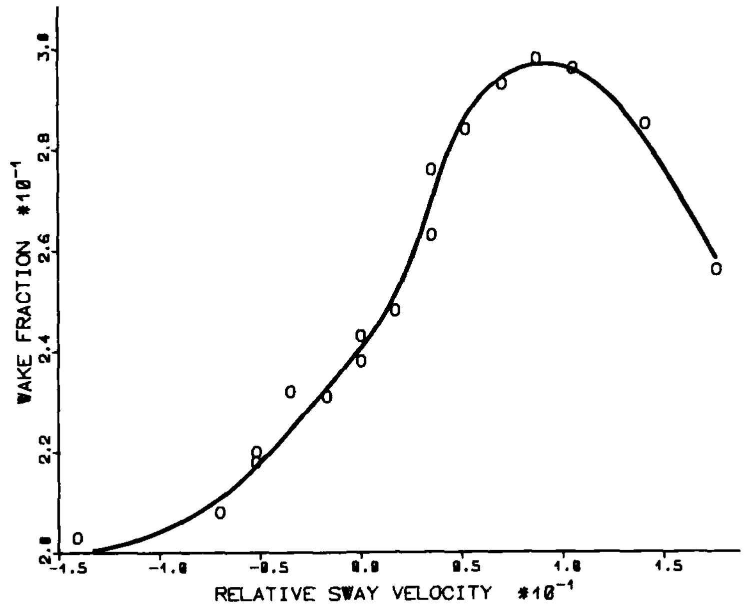


Fig 3. Wake fraction W(r,v) plotted over relative sway velocity at the stern v_p/u for a Mariner type hull. From Jørgensen and Prohaska [7].

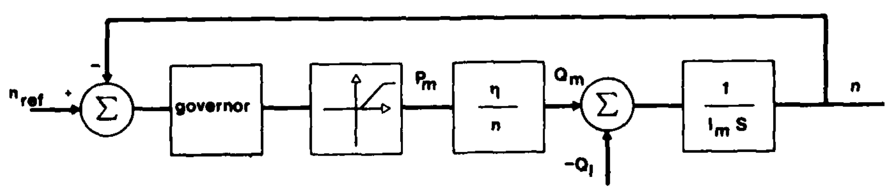


Fig 4. Structure of general engine model.

accounts for the effects of propeller loading when expressed as follows [8]

$$\bar{c}^2 = \alpha(V_A + k_R U_A)^2 + (1-\alpha)\, V_A^2 \qquad (12)$$

where

α is the ratio between screw diameter and height of rudder

$k_R U_A$ is the propeller induced velocity at the rudder

The term U_A is the theoretical screw induced velocity infinitely downstream and is expressed from momentum theory by

$$U_A = \sqrt{V_A^2 + \frac{8}{\pi}\frac{T}{\rho D^2}} - V_A \qquad (13)$$

Model tests reveal the coefficient k_R to equal unity for practical purposes, meaning the propeller slipstream has nearly reached its maximum value of U_A+V_A at the rudder. Thus the mean flow $\bar{c}$, from eq. (12) and (13), simplifies as

$$\bar{c}^2 = V_A^2 + \frac{8\alpha}{\pi\rho D^2}\, T \qquad (14)$$

$$= V_A^2 + c_T^2\, T$$

The combined performance of screw, rudder and engine is completed when the dynamics of the engine is included.

Engine Equation. From eq. (5) it is apparent that the engine model required must generate the screw angular velocity according to some function of command inputs and in turn generate a torque to compensate for the propeller load. The model has to be accurate only to the extent it copies dynamic response when subject to load-variations and command settings. Formulation in terms of energy balance then is adequate for adaption to the different types of engines and associated control mechanisms.

Power inlet to the motor P_m is proportional to fuel flow, and power outlet equals developed torque Q_m times screw angular velocity n. Further any difference between developed torque Q_m and load torque will change the timerate of screw revolutions. The basic equations thus are

$$Q_m n = \eta P_m \qquad (15)$$

$$I_m \dot{n} = n Q_m - n Q_1$$

where I_m is the total inertia of rotating parts plus added inertia from the water as referred to the shaft. η is the efficiency of the

motor.

The build up of developed torque from power inlet reflects the dynamics of the motor. In eq. 15 this is taken as instantaneous because the engine response is very fast compared to the remainder dynamics of the ship.

The governor control loop contributes by its low frequency performance which determines the stiffness of RPM control to load variations. In fig. 4 a block diagram of the general engine model is shown.

The Speed Loss Equation. From the preceeding sections the dynamical speed loss equation is formulated for a vessel without external disturbances acting upon it.

$$
\begin{aligned}
(m-Y_{\dot{v}})\dot{v} &= (Y_{\dot{r}}-mx_G)\dot{r}+(Y_{ur}-m)ru+Y_{uv}uv+Y_{cc\delta}\bar{c}^2\delta \\
(I_2-N_{\dot{r}})\dot{r} &= (N_{\dot{v}}-mx_G)\dot{v}+(N_{ur}-mx_G)ru+N_{uv}uv+N_{cc\delta}\bar{c}^2\delta \\
m\dot{u} &= x_{uu}u^2+(x_{vr}+m)vr+(x_{rr}+mx_G)r^2+x_{cc\delta\delta}\bar{c}^2\delta^2+(1-t)T \\
I_m\dot{n} &= Q_m-Q_l \\
T &= T_{nn}n^2+T_{nv}nV_A \\
Q_l &= Q_{nn}n^2+Q_{nv}nV_A \\
V_A &= (1-w_o)u-w_v(v-l_{PG}r)u \\
\bar{c}^2 &= V_A^2 + c_T^2\,T \\
Q_m &= \eta\, P_m
\end{aligned}
\tag{16}
$$

The $\dot{v}$ and $\dot{r}$ relations show the well known dynamical equations of motion and those of $\dot{u}$ and $\dot{n}$ constitute the propulsion dynamics. The remaining expressions form the algebraic bounds brought up by screw and wake behaviour.

The upper two sway and yaw rate equations in (16) have the rudder angle as driving element. Speed and torque dependancy is affecting the coefficients only. Therefore, this part describing the hull dynamics is decoupled from the speed loss equations for the small excursions from steady advance considered. The sway velocity and the rate of turn are then literally inputs to the speed equation although they are generated by antecedent rudder angle movements.

External Disturbances. Excitations on a vessel from waves and wind are complicated functions of the shape of hull and superstructure and of the angle of attack. When forces in the motional equation vary with the ships heading, the course becomes necessarily a state in the equations, and the external disturbances then establish feedback paths from the course to yaw and sway accelerations. These time varying coefficients have not been included in this exposition because they do not affect the speed loss equation. Further the full scale trials were all carried out in nice weather.

When autopilot performance is considered a model is adapted, which utilizes a set of transferfunctions relating disturbance power density with power density of the resulting movement. The excitation then follows from the spectrum of the actual disturbance passing the complex transferfunction. The spectrum of wind pressure is approximately white whereas the wave spectrum has bandpass nature. When the ship meets the disturbances at steady advance velocity, the wavespectrum is transformed by a nonlinear convolution due to the natural dispersion of ocean waves. In following sea this effect can concentrate major wave energy in a tiny bandwidth at a fairly low frequency of encounter. This situation calls for specific attention when basic autopilot performance is investigated. A plot of the encounter spectrum is shown in fig.5.

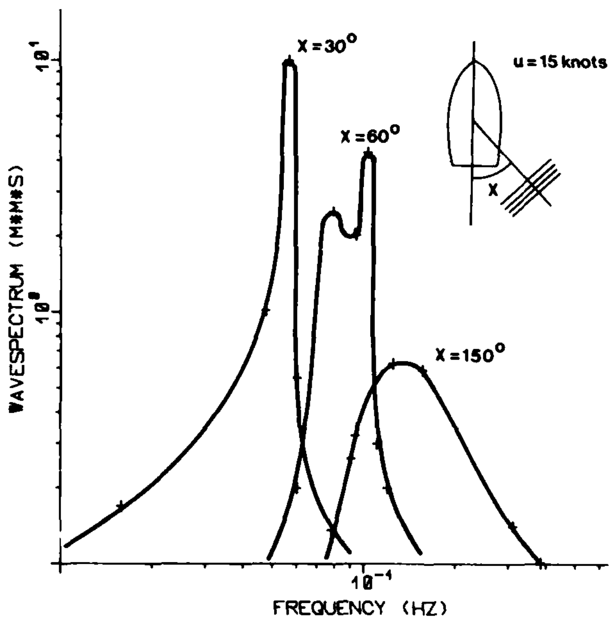


Fig.5 Wave spectrum as seen from ship at different angles of attack.

Summary of Ship Dynamics.

In this chapter a model of a vessel has been outlined which includes the effects of propeller interaction with hull movements through dynamical variations of the wake. A simple input - output model of the engine completed the description as to form the present model of the speed loss dynamics.

TECHNIQUES OF SYSTEM IDENTIFICATION.

Application of modern identification techniques has been a fast growing field of practical exercise ever since efficient mathematical methods and proven computer algorithms were developed some years ago. Methods of least squares fitting to curves in the phase plane and different methods of frequency and time response analysis has a tradition of long practical experience,when deterministic signals are involved. The novel methods in contrast offer the possibility of quantifying stochastic disturbances also in a mathematical framework which is well posed for applications of modern control theory.

At the time of planning of the experiments, however, maximum likelihood methods had not proven efficient when nonlinear systems were involved and were still at a state of active research. Actually, the first applications to nonlinear ship steering dynamics were recently reported [11].

Instead frequency analysis techniques were considered. They require the system to be excitated by a signal having its major energy at one or more distinct frequencies. In the case of the nonlinear speed equation, excitation by a single sinewave in the rudder angle or rate of turn holds a unique achievement. A linear term in the equation will respond at the excitation frequency whereas a square term will produce a response containing double the excitation frequency. Thereby different nonlinearities appear as transparent effects. Furthermore, methods of spectral analysis are well fitted to depict the very small amplitude loss terms of interest.

The frequency analysis method does not allow for the sophisticated automatic validation of model structure as does the system identification approach. Instead the physical comprehension as reflected in eq. (16) has to be relied on and a more laborious effort is required to validate the model.

Reformulation of Speed Loss Equation. The structure of the speed equation must be changed from eq. (16) to an explicit input output form when used for identification purposes.
The sway and rate of turn equations combine to form the two transferfunctions below, both having rudder angle as input.

$$\frac{r(s)}{\delta(s)} = c_r \frac{1+s\tau_r}{(1+s\tau_1)(1+s\tau_2)} \tag{17}$$

and

$$\frac{v(s)}{\delta(s)} = c_v \frac{1+s\tau_v}{(1+s\tau_1)(1+s\tau_2)} \tag{18}$$

The timeconstants and gain factors are easily derived from eq. (16) and has been given in the literature.

The speed loss equation is mostly truncated to include the rv and the δ^2 loss terms only because RPM control is assumed to be ideal In actual vessels, the governor controller has a certain stiffnes to

load variations and variations in RPM are induced. Combined with the antisymmetric wake variation with sway velocity at the stern, this can contribute significantly to propulsion losses. This is seen by expanding the speed equation from steady state.

By elaborating eq. (16) using the notation U_o, n_o, w_o and V_o for the steady state values of ship speed, screw revolutions, wake and inflow velocity at the propeller, the set of small signal relations eq. (19) are obtained. The symbols u and n now denote the deviations of speed and screw rotation from the steady state values. ΔQ_m is differential torque and G(n(t)) denotes governor response.

$$\begin{aligned} m\dot{u} &= 2X_{uu}U_o u+(X_{vr}+m)vr+X_{cc\delta\delta}(1-w_o)^2U_o^2\delta^2+(1-t)T_{nn}n^2 \\ &\quad +(1-t)(2T_{nn}n_o n+T_{nv}n_o(1-w_o)u+T_{nv}U_o w_v n(v+l_{PG}r)) \\ I_m\dot{n} &= \Delta Q_m-2Q_{nn}n_o n+Q_{nv}n_o U_o w_v(v-l_{PG}r) \\ &\quad -Q_{nv}n_o(1-w_o)u-Q_{nv}U_o(1-w_o)n \\ \Delta Q_m &= G(n(t)) \end{aligned} \tag{19}$$

A block diagram showing the structure of the small signal speed loss equation is shown in fig. 6 . Propeller revolution is seen to be excitated by the stern sway velocity and the resulting variation in RPM gives a net drag force proportional to $n(v-l_{PG}r)$. The magnitude of this term is discussed below.

By neglecting the second order influence on speed and RPM from the propulsion loss, the transferfunctions listed in eq.(20) result. Expressions for timeconstants and gain factors in eq. (20) are listed in table 1.

$$\begin{aligned} \frac{n}{(v+l_{PG}r)} &= k_v k_m \frac{(1+s\tau_s)}{(s\tau_m+1)(s\tau_s+1)-k_m k_s a_1 a_2+(s\tau_s+1)G(s)k_m} \\ \frac{u}{T_{loss}} &= k_s \frac{k_m G(s)+(s\tau_m+1)}{(s\tau_m+1)(s\tau_s+1)-k_m k_s a_1 a_2+(s\tau_s+1)G(s)k_m} \end{aligned} \tag{20}$$

Total drag force from the loss terms is finally summarized in eq. 21.

$$\begin{aligned} T_{loss} &= (x_{vr}+m)vr+x_{cc\delta\delta}(1-w_o)^2U_o^2\delta^2 + \\ &\quad (1-t)(T_{nn}n^2+T_{nv}U_o w_v n(v+l_{PG}r)) \end{aligned} \tag{21}$$

In the transferfunctions eq. (20) G(s) is the transferfunction of the governor, from RPM deviation from its setpoint to developed motor torque. This controller normally includes a pure integration, and the two transferfunctions are generally of third order.

Term	Expression	Magnitude	Unit
k_v	$w_v \cdot Q_{nv} n_o U_o$	$\sim 4 \cdot 10^5$	Nms/m
k_m	$1/(2Q_{nn} n_o + Q_{nv} U_o (1-w))$	$3 \cdot 10^{-6}$	rad/Nms
k_s	$-1/(2x_{uu} U_o + (1-t) T_{nv} n_o (1-w_o))$	$5 \cdot 10^{-6}$	m/Ns
a_1	$-Q_{nv} n_o (1-w_o)$	$1.3 \cdot 10^5$	Nms/m
a_2	$(1-t) \cdot 2T_{nv} n_o (1-w_o)$	$-2.3 \cdot 10^5$	Ns/rad
τ_m	$I_m/(2Q_{nn} n_o + Q_{nv} U_o (1-w_o))$	1.5	sec
τ_s	$-m/(2x_{uu} U_o + (1-t) T_{nv} n_o (1-w_o))$	100	sec

Table 1. Expressions for timeconstants and gain factors in the speed equation.

Term	Magnitude	Unit	Term	Magnitude	Unit
V_A	9.0	m/s	$T_{nn} n_o^2$	$2.5 \cdot 10^6$	N
n_o	10.0	rad/s	$T_{nv} n_o V_A$	$-1.9 \cdot 10^6$	N
T_{nn}	$2.1 \cdot 10^4$	Ns^2/rad^2	T	$0.6 \cdot 10^6$	N
T_{nv}	$-1.8 \cdot 10^4$	$Ns^2/m \cdot rad$	$Q_{nn} n_o^2$	$2.5 \cdot 10^6$	Nm
Q_{nn}	$2.1 \cdot 10^4$	Nms^2/rad^2	$Q_{nv} n_o V_A$	$-1.6 \cdot 10^6$	Nm
Q_{nv}	$-1.6 \cdot 10^4$	$Nms^2/rad\ m$	Q	$0.9 \cdot 10^6$	Nm
$w_r \cdot U_o$	0.4-3.0		x_{uu}/m	$-0.2 \cdot 10^{-3}$	m^{-1}
w_o	0.20		x_{vr}/m	0.5	
(1-t)	0.8		$x_{cc\delta\delta} V_A^2/T$	-1.2	$m/s^2 \cdot rad^2$

Table 2. Typical set of data for a cargo ship.

Identificability and Techniques of Analysis. Provided measurements of sway velocity and rate of turn are available the transfer-function $n/(v-l_{PG}r)$ can be derived from measurements in terms of values of poles zeros and static gain factor. These are composite expressions of the hydrodynamic coefficients and a simple analysis shows that they cannot individually be obtained. The slope of the wake variation is not obtainable either. By combining excitations from engine torque with disturbances in v and r from rudder movements however, the effective coupling k_v from sway velocity at stern to load torque variation can be determined.

The drag force from losses must be computed from the different squareterms and crossproducts δ^2, n^2, vr and $n(v-l_{PG}r)$. When hull movements are brought about solely by rudder fluctuations, v and r relate to δ through eqs. (17) and (18). As further n relates to v and r through eq. (20) it is impossible to identify the loss term coefficients from eqs. (21) and (20) directly. By using uncorrelated excitations from rudder and engine torque instead, the coefficients are in theory identificable when the dynamic relations between the variables are also taken into account.

In the actual experiments only rudder movements were used to create hull motions because of hardware constraints. Numerical values of the individual loss term coefficients are therefore not attained but the magnitude of speed loss generation can be verified.

When the excitation signal is a single frequency sinewave, drag force from the loss appears as second harmonic of the input frequency in the speed recording. Instantaneously thrust oscillations at the input frequency occur from the variations in wake. Because of the tight coupling of the system, according to fig.6 , a calibration of the speed reading is thus provided.

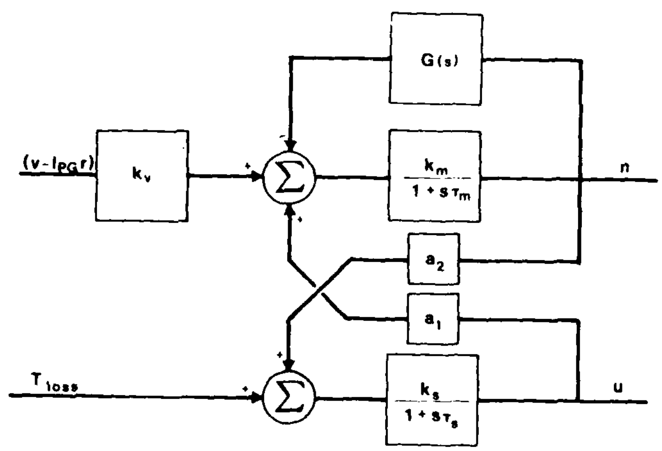


Fig.6. Small signal speed loss equation.

ELEMENTS OF AN INTEGRATED CONTROL SYSTEM FOR LARGE SURFACE EFFECT SHIPS (SES)

by Warren L. Malone
Naval Sea Systems Command (PMS 304)
Washington, D.C. 20084

ABSTRACT

This paper discusses some of the approaches that have been considered for use in the control of large SES, the general arrangement of the ship control system, the Ride Control System (RCS), and the Maneuvering System.

The algorithms considered for the more critical functions of maneuvering control and heave control are discussed and the current treatment of the control allocation law is presented.

INTRODUCTION

The majority of advanced ship designs place at least some emphasize on high speed operation and reduction of manning. This trend has resulted in technology programs to provide a greater level of automation in all aspects of ship control. In addition, the marked increases in availability and reliability of data processing equipment have made the application of this equipment highly competitive with more traditional techniques. The design of a large SES thus provides a timely opportunity for application of some of this new technology.

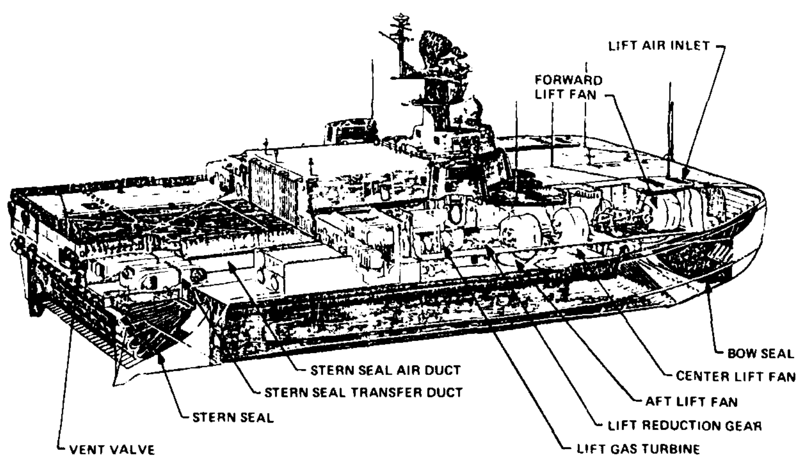


Figure 1. The general configuration of a large SES.

11. Norrbin, N. H., Åström, K. J., Byström, L., Källström, C. G.
Further Studies of Parameter Identification of Linear and Nonlinear Ship Steering Dynamics.
SSPA report 1920-6, October 1977.

12. Abkowitz, M. A.
System Identification Techniques For Ship Manoeuvring Trials.
Office of Naval Research Control Conference, Monterey 1975.

13. Nørtoft Thomsen, J. C.
Analogsteuerung.
Hansa II. Sept.-heft 1976 nr. 18, pp. 1538 - 1540.

14. Huang, T. T., Wang, H. T., Santelli, N., Groves, N. C.
Propeller-Stern-Boundary Layer Interaction on Axisymmetric Bodies: Theory and Experiment.
David W. Taylor Naval Ship R and D center, December 1976 pp. 76.

15. Zuidweg, J. K.
Automatic Guidance of Ships as a Control Problem Thesis.
Delft T. H., 1970.

16. Bech, M. I.
Some Aspects of the Stability of Automatic Course Control of Ships.
Journal Mech. Eng. Science, vol. 14 no. 7, 1972.

17. Koyama, T.
On the Optimum Automatic Steering System of Ships at Sea.
Soc. of Naval Arch. of Japan. Vol. 122, 1967, pp. 142-145.

18. Andersen, T. E.
On Dynamics of Large Ship Diesel Engines. Thesis.
Control Engineering Laboratory, Technical University of Denmark, June 1974.

REFERENCES

1. Abkowitz, M. A.
   Stability and Motion Control of Ocean Vehicles.
   MIT Press, 1972, sec. ed.

2. Norrbin, N. H.
   Theory and Observations on the use of a Mathematical Model for Ship Manoeuvring in Deep and Confined Waters.
   SSPA publications no. 68, 1971.

3. Bishop, R. E. D., Burcher, R. K., Price, W. G.
   The Determination of Ship Manoeuvring Characteristics from Model Tests.
   The Royal Institution of Naval Architects, vol. 117, 1975 pp.215-231.

4. Nomoto, K. and Motoyama, T.
   Loss of Propulsion Power caused by Yawing with Particular Reference to Automatic Steering.
   I. Soc. Nav. Arch. Japan 120 (1960 p. 71)
   BSRA Translation No. 2610.

5. Strøm-Tejsen,J., Chislett, M. S.
   A Model Testing Technique and Method of Analysis for the Production of Steering and Manoeuvring Qualities of Surface Vessels.
   Danish Ship Research Laboratory, HY-7, 1966.

6. Bech, M. I., Smitt, L. W.
   Analog Simulation of Ship Manoeuvring based on full-scale trials or free-sailing model tests.
   HyA Report Hy-14, Sept. 1969.

7. Jørgensen, H. D.,Prohaska, C. W.
   Wind Resistance,Report of Performance Committee, App. IV.
   Proc. of 11th ITTC, Tokyo, 1966, p. 123.

8. Berlekom, W. B., van.
   Effects of Propeller Loading on Rudder Efficiency.
   Fourth Ship Control System Symp.
   Haag, 1975, pp. 5.83 - 5.98.

9. Åström, K. J., Källström, C. G.
   Identification of Ship Steering Dynamics.
   Automatica, vol. 12, 1976, pp. 9 - 22.

10. Åström, K. J., Källström, C. G., Norrbin, N. H., Byström, L.
    The Identification of Linear Steering Dynamics Using Maximum Likelihood Parameter Estimation.
    SSPA publications no. 75, 1975.

CONCLUSION

From the mathematical description of ship forward speed dynamics it was found that variations in wake fraction due to varying direction of inflow to the propeller cause fluctuations in RPM, and also gives rise to a drag force component

$$T_{nv}U_o w_v n(v+l_{PG}r)$$

From the full scale experiments it is demonstrated that the contribution from this effect can be significant and certainly when autopilots are considered, which possess very low steering generated drag force. Coordination of RPM control with course control therefore seems advantageous.

The identification of the nonlinear speed equation by means of frequency analysis techniques showed successful in validating the structure of the mathematical model and in determining a set of parameters in the input-output description of the system. The values found compare reasonably to those predicted. Explicit determination of the individual loss terms was not possible from the experimental data for two reasons. Only excitations from the rudder were used in the experiment,and the exploration of the dynamics of one decade in frequency is hardly enough for the identification.

Finally hydrodynamic effects appeared as an essential element of the dynamic wake variation.

ACKNOWLEDGEMENT.

Grateful acknowledgement is due to the staff of the Hydrodynamics group at the Danish Ship Research Laboratory, to professor J.R.Jensen at the Servolaboratory and to Mr. J. C. Nørtoft-Thomsen and Mr. M.I. Bech for the inspiration and advice gained through the study.

The author is indebted to the Danish Shipping Company A.P.Møller for providing the data from the full scale experiments.

AUTOPILOT PERFORMANCE

The performance of course keeping autopilots regarding the speed loss from steering activity is determined through its response to disturbances from waves and wind.

The excitation of the hull from waves can be calculated from strip theory methods, and computer programs excist that calculate amplitude and relative phase of the disturbed motions expressed in sway velocity and yaw rate. They add to v and r generated by the rudder, to form the resulting hull motions. The frequency spectra of the disturbances have a form similar to the wave spectra in fig. 5. In following sea the excitations gather at a relatively low frequency, which is normally somewhat higher than the closed loop bandwidth of the system. As rudder movements have little effect in that frequency range, the effect of wave interaction is mainly to cause rudder generated speed loss. The vr generated speed loss is not necessarily excessive when large excursions in v and r are brought about by waves, because the phasing of v and r in some situations of following sea are in quadrature. The rudder induced drag in this situation will be proportional to the square of the high frequency gain of the autopilot and considerable savings are thus gained by decreasing the system bandwidth.

The contribution to speed loss from the wake generated variations in RPM is best judged from an example.

Example. Assume that a 0.2 deg/sec rate of turn and 0.5 m/s sway velocity is generated in quadrature by waves. Local sway velocity at stern then is approximately of 0.6 m amplitude and 30° phase. From the estimated value of k_v, this gives an amplitude in RPM of 0.3 rad/sec. Note that from the n/r plot the governor will not suppress the variations in the range of frequencies relevant to following sea excitations. From table 2 and 3, the drag force is then found to be

$$T_{nv} \cdot U_o \cdot w_v \cdot n(v+l_{PG}r) = 8 \cdot 10^3 N$$

as total thrust is $0.6 \cdot 10^6$ N, this is 1.3%.

The wake induced loss is thus significant when optimal performing autopilots are considered. Coupling of the RPM controller to the course keeping autopilot therefore seems to be advantageous as to attempt proper phasing of torque from the engine and steering activity in order to minimize propulsion losses.

Instead, from the r/δ function, the magnitude of τ_1 was obtained, and the remaining parameters in the n/r and u/r functions were then estimated, assuming the value of τ_1 as fixed. This did give consistent results that show good agreement with the theoretical predictions. The estimated parameters are listed in table 4.

The accuracy of the log reading is of special concern. By using the relation between u/r and n/r, one gets a prediction of the gain in the u/r function of 102 m/rad, based on the measured n/r gain estimate. The estimation of u/r gives 128 m/rad. The logreading is therefore relied on although some spread in the measurements are apparent. The outlier in the u/r plot originates from an experiment having a very large amplitude in yaw rate. Influence from the boundary layer is presumably causing the misreading in this case.

On the plots of u/r and n/r the measured phase is seen to have different shape than that expected from the transferfunctions in table 3. The measured phase has an ever decreasing asymptote instead of approaching a limited value as expected. This is explained by the presence of a timedelay in the build up of wake from rate of turn. A computation of the delay gives consistent values for the transferfunctions u/r and n/r, and also for u/δ^2. In the speed equation therefore k_v must be replaced by $k_v \cdot e^{-s\tau}$. This result is not quite as significant from the n/r measurements as from u/r due to spread of the high frequency values. It is accepted due to the structure of the system, which is verified by the measurements anyway.

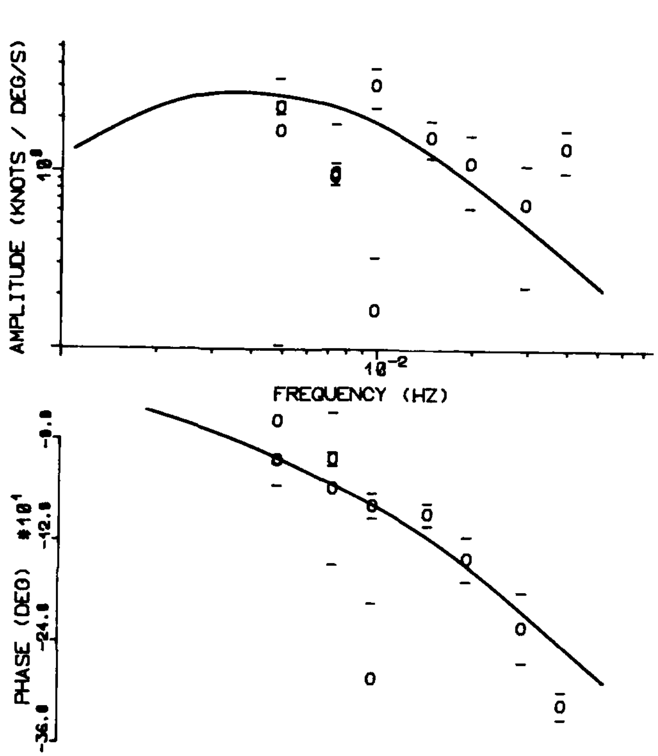
Hydrodynamic memory effects are described in the literature [3], and are known to give rise to timedelays. The magnitude was surprising, however, and even more as the r/δ measurements bear no indications of memory effects.

Validation of the loss term equation is difficult because of its complexity. At low frequencies the rv, nr and n^2 terms will dominate, and at high frequency rudder induced loss is preponderant. The frequency spacing of the doutle pole in $1/\tau_r$ to the pair of zeros, which depend on the magnitudes of x_{vr}, $x_{\delta\delta}$, T_{nv} etc., determine the relative contributions of the losses. In fig. 12 an characteristic is drawn, which approximates the measured values. It has the poles according to table 3 and two zeros 0.15 Hz. The position of the zeros indicates the ratio of the x_{vr} and $x_{\delta\delta}$ loss terms to be of magnitude 30. This is not too far from values obtained from model tests, but the result is not significant. The DC value of 3 knots/degree of rudder is also much larger than expected from practical experience.

Results of measurements.

It has been demonstrated that the use of frequency analysis is feasible to verify the structure of the speed loss equation, and that values of comparable magnitudes to those anticipated are obtained. The measurements demonstrate that the effects of dynamic wake variations on the full scale vessel are perceptible to RPM and shaft torque. Further the presence of memory effects in the build up of wake from hull motions is rendered.

The experiments fail to reveal magnitudes of the different loss terms, but values of entire speed reduction from drag forces are obtained. The speed loss due to wake variation is assessed by using the measured value of k_v together with thrust coefficients of the screw.



Dif. 11. The transferfunction $u(s)/r(s)$

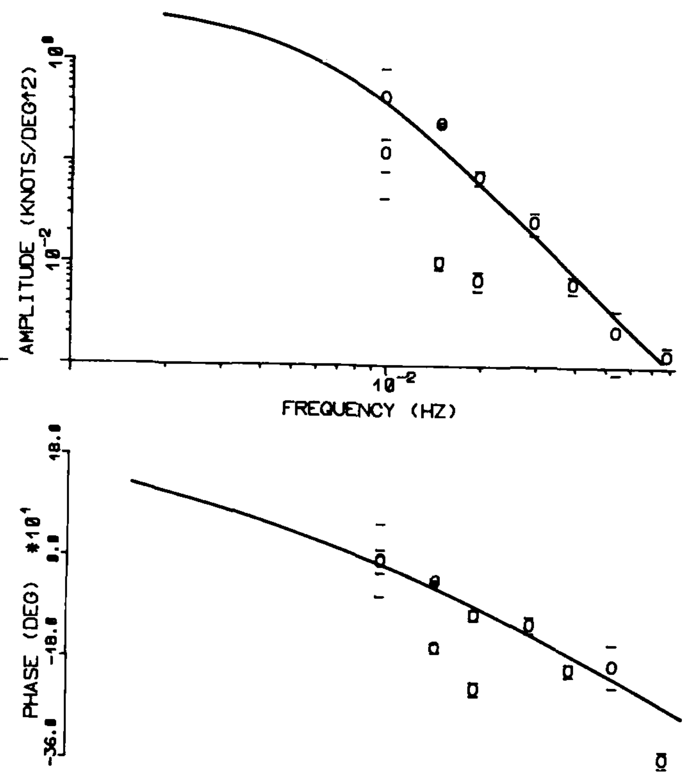


Fig. 12. The transferfunction $u(s)/\delta^2(s)$

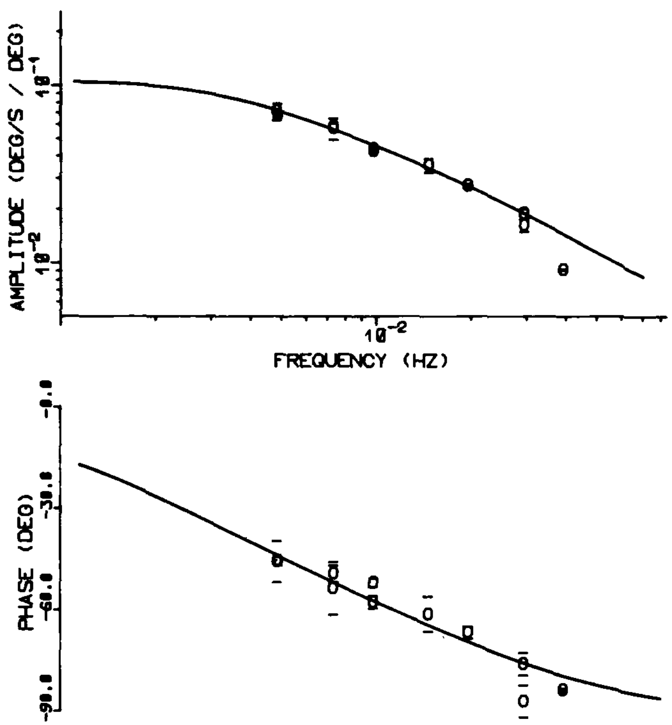


Fig. 9. The transferfunction $r(s)/\delta(s)$

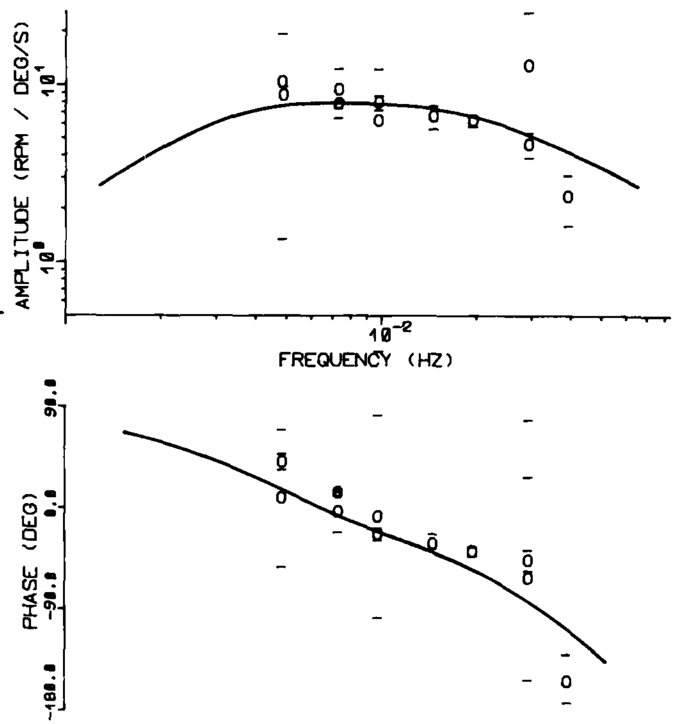


Fig. 10. The transferfunction $n(s)/r(s)$

Transferfunction	Reduced expression
$\frac{r}{\delta}\frac{(s)}{(s)}$	$c_r \frac{1+s\tau_r}{(s\tau_1+1)(s\tau_2+1)}$
$\frac{n}{r}\frac{(s)}{(s)}$	$k_v k_m l^* \frac{s}{s+k_G k_m} \cdot \frac{1+s\tau^*}{1+s\tau_r}$
$\frac{u}{r}\frac{(s)}{(s)}$	$k_v k_m a_2 k_s \frac{s}{s+k_G k_m} \cdot \frac{1+s\tau^*}{1+s\tau_r} \cdot \frac{1}{1+s\tau_s}$
$\frac{u(s)}{\delta^2(s)}$	$\frac{k_s}{1+s\tau_s} \left(\frac{c_v}{c_r} x_{vr} c_r^2 \left(\frac{1}{1+s\tau_1}\right)^2 + x_{\delta\delta} + \right.$ $(1-t)T_{nn} \cdot \left(\frac{n(s)}{r(s)} \frac{r(s)}{\delta(s)}\right)^2 +$ $\left. (1-t)T_{nv}U_o w_v \frac{n(s)}{r(s)} \frac{(v-l_{PG}r)}{r(s)} \left(\frac{r(s)}{\delta(s)}\right)^2 \right)$

Table 3. Reduced order transferfunctions.

r/δ		n/r		u/r	
c_r	0.109 s^{-1}	$k_v k_m e^*$	89	$k_m k_v a_2 k_s l^*$	128 m/rad
τ_r	25.1 s	$1/k_m k_g$	12.7 s	τ_s	~ 49 s
τ_1	54.1 s	τ^*	2.4 s		
τ_2	13.8 s				
		Delay	8.0 s	Delay	8.0 s

Table 4. Results of frequency analysis.

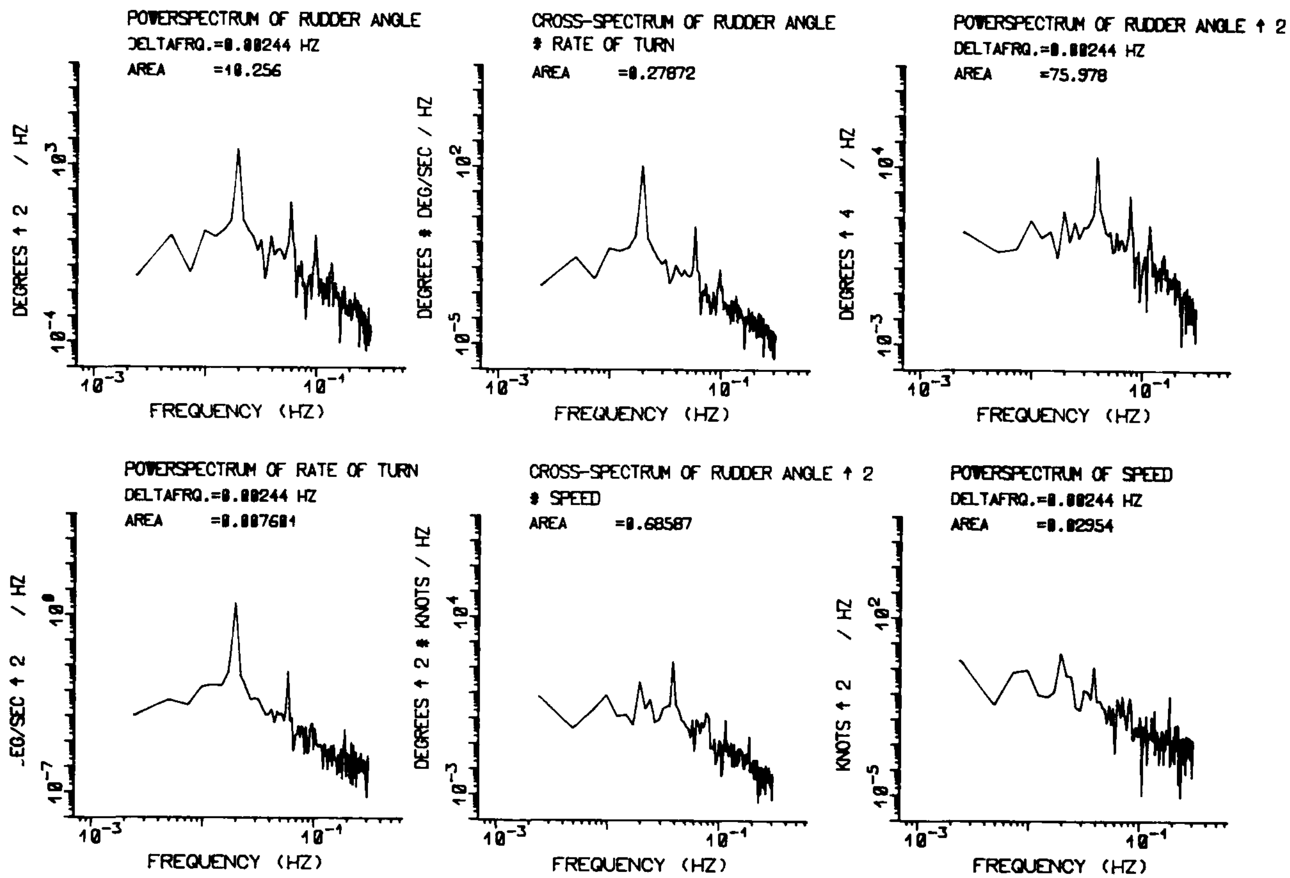


Fig. 8. Power spectra and amplitude parts of cross spectra from oscillating experiment.

Data Analysis.

From the recordings, various powerspectra and crossspectra have been computed to obtain the transferfunctions of interest. The spectral analysis was carried out using fast fourier transform techniques together with application of data smoothing in order to obtain minimal resulting variance.

For the problem at hand simply averaging transforms of consecutive timeslices showed advantageous.

In fig. 8 some characteristic spectra are plotted. In the particular run, excitation frequency is 0.02 Hz. The spectra of rudder angle and rate of turn exhibit excellent signal to noise ratios as also apparent from the plots of timehistory. In the spectrum of the rudder angle some 3. order and higher harmoni s are present. This is due to slow rate saturation of the rudder which can also be seen from the nearby triangular shape on the recording of rudder angle over time.

The most interesting spectrum is that of the speed. It clearly contains significant power at both the excitation frequency from the wake variation and at double the excitation frequency due to the loss terms. Signal to noise ratio of the speed measurement is not impressive but it is sufficient to obtain significant results when computing transferfunctions.

The transferfunction from rudder angle squared to second harmonic of measured speed constitutes the input-output model of the speed loss, when only excitation from the rudder is considered. Values of this transferfunction are obtained from the cross-spectra between δ^2 and u and the powerspectra of the square of δ. The values can then be compared with the theoretical expectation obtained from the previous equations.

By direct substitution into eq. (20) and (21) high order expressions result which are irrelevant to the range of frequencies of interest . Some simplifications are possible, however, to achieve reduced order relations. They are

- The engine timeconstant $\tau_m << \tau_s$ thus $\tau_m \sim 0.0$
- The governor is approximated by a pure integration $G(s) \sim k_g/s$
- The timeconstants τ_1 and τ_2 are considered to cancel in the r/δ transferfunction when u/δ^2 is computed.

The set of reduced order transferfunctions listed in table 3 are those used for the model validation. Note that the n/r expression contain a pure differentiation due to the integrating part of the governor.

The results obtained are plotted in fig. 9 to 12. Each computed value is circled by an O and the 0.95 confidence limits of each is marked by bars.

It was first attempted to estimate the transferfunctions from least squares fitting to each set of data. This was not successful. The complexity of the transferfunctions and the spread of measurements gave inconsistent values of poles and zeros.

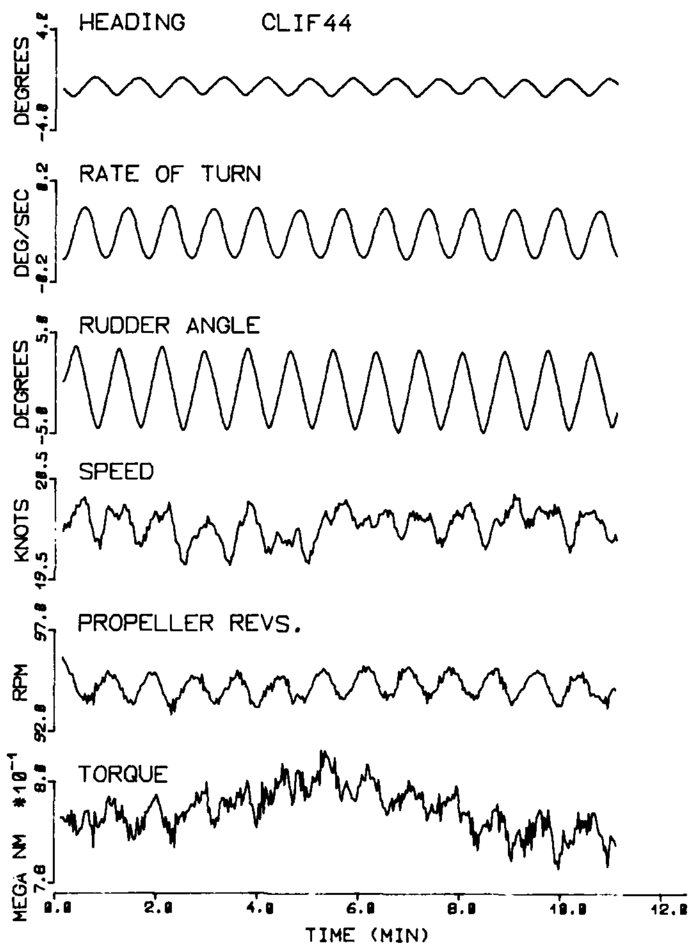


Fig. 7. Recordings from oscillating experiment.

THE CLIFFORD MAERSK EXPERIMENT

The frequency analysis technique described in the previous sections has been used on a container ship Clifford Mærsk owned by the Danish shipping company A. P. Møller. A number of experiments were conducted where the vessel was oscillated about a straight course by means of the rudder. The wavy course was brought about by a sinewave signal added to the course error voltage inside the autopilot.

Experimental setup.

The Clifford Mærsk is equipped with a diesel engine having RPM controlled by a governor. For achieving information on values of the control inputs and output signals required for the identification, the following instrumentation was used:

- ψ course from gyro compass
- r rate of turn from rate gyro
- δ rudder angle from potentiometer at the rudder stock
- n screw revolutions from tachometer
- u from pitot tube log located near the stern
- Q_m not directly measured but an index setting sensor was mounted and the shaft torque was obtained through calibration.

The experiments were all carried out in calm water in order not to complicate the data analysis from external disturbancies. Measurements were made at various frequencies between 0.005 Hz and 0.04 Hz, the range which is most important to the manoeuvring dynamics of the particular vessel. The range was chosen from practical reasons also, however. At lower frequencies run time of each experiment necessarily becomes excessive, and above 0.04 Hz the rudder will not respond reasonably to large commands due to slew rate saturation.

A sample of the timehistory of one experiment is plotted in fig. 7.

A measurement of sway velocity v was not available. Instead the similarity between r, the rate of turn and v must be used to eliminate v from the transferfunctions, eq. 20. When the hull motion is due to rudder activity eqs. (17) and (18) combine to express v from r as

$$v = \frac{c_v}{c_r} \cdot \frac{1+s\tau_v}{1+s\tau_r} \cdot r \tag{24}$$

The sway velocity at the stern is then related to r through

$$v - l_{PG} r = -(l_{PG} - \frac{c_v}{c_r}) \frac{1+s((l_{PG}\tau_r - c_v\tau_r/c_r)/(l_{PG} - c_v/c_r))}{1+s\tau_r} r \tag{25}$$

The constants in eq. 25 are all positive because c_v is negative due to the sign convention. In the next section the gain in eq. (25) is referred to as l^* and the numerator timeconstant is abbreviated τ^*.

Estimation of Transferfunctions. Digital time series analysis by means of frequency domain techniques is well established, and various methods excist for estimating transferfunctions for a system from recordings of inputs and outputs.

If the powerspectrum of the input x is $G_x(f)$, where f is the frequency, and the transferfunction of the system is H(f), then the powerspectrum of the output y will be $G_y(f)$.

$$G_y(f) = |H(f)|^2 \, G_x(f) \tag{22}$$

The spectrum of the cross correlation function between x(t) and y(t) is $G_{xy}(f)$.

$$G_{xy}(f) = H(f) \cdot G_x(f) \tag{23}$$

The powerspectra are real valued functions and contain information on amplitude only. The cross spectrum is complex valued, and from (23) $G_{xy}(f)$ is seen to have its imaginary value equal to that of the system. By means of $G_{xy}(f)$ and $G_x(f)$ it is thus possible to obtain both amplitude and phase of the transferfunction H(f). The redundant information on the amplitude $|H(f)|$ when $G_y(f)$ is also available is used to establish the confidence of the measurement. In the plots of transferfunctions in the next chapter, the intervals of confidence level 0.95 are indicated. The likelihood is 95% that each value is within the intervals.

Here fast fourier transform techniques were used to compute power and cross spectral densities,and data smoothing methods were carefully tested because very optimistic confidence intervals may result due to bias effects.

Summary of Identification Methods.

In this chapter frequency analysis methods have been discussed and found attractive regarding the separation of the linear from the nonlinear contributions to speed variation. The speed equation was expanded from a steady state assuming small signal deviations and thereby a set of input-output relations was obtained. The consecutive discussion of identificability concluded that the full set of coefficients can not be determined when only excitations from the rudder are used.

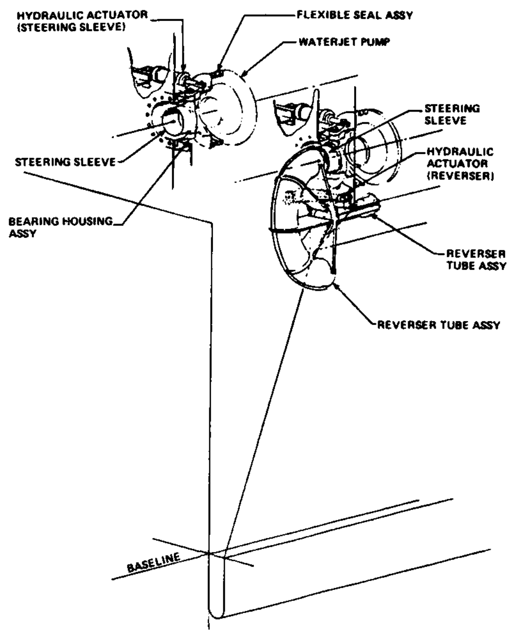


Figure 2. Relationship of the main reverser tube assembly to the inboard and outboard waterjet steering sleeves.

A large SES, originally planned for 2000 tons but currently fixed at 3000 tons, has been under study for some time. A variety of designs have been considered but most versions have had the general configuration indicated in Figure 1. The ship is supported on a cushion of air confined within the plenum defined by the ship's two side walls, its bow and stern seals, and the sea surface. The air mass for the plenum and for the bow and stern seals is supplied from gas turbine driven centrifugal fans and is distributed via air ducts positioned across the width of the ship. Propulsive drive is supplied by four stern mounted waterjets whose inlets are located on the lowermost portion of the respective sidewalls. The waterjets are vectored (Figure 2) by steering sleeves mounted at the waterjet exits. Reversing forces are generated by bringing "U" shaped tubes called reversing tubes into place in front of the respective outboard waterjets.

The SES design includes provisions for both local and remote control of the bow and stern seals, the lift system engines, the "reversers," and the propulsion system, including vectoring of the waterjet nozzles. The interconnection network for the remote control is a distributed data system of the form indicated in Figure 3. The system consists of local data terminals which relay data between two control stations, the Ship Control Console (SCC) located in the pilot house and the Central Control Station (CCS) located in close proximity to the SCC but one deck below.

The functional split between the SCC and CCS is shown schematically in Figure 4. The various operational and ship plant monitoring signals are picked up by the closest data terminal (with the exception that certain critical signals are placed on separate data terminals to preclude total loss of ship capability through loss of a single

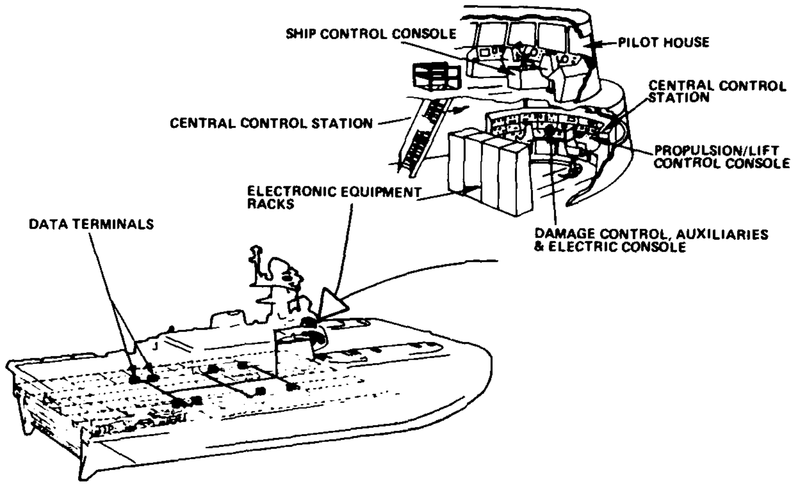


Figure 3. Location of the Ship Control Console, Central Control Station, and Data Terminals.

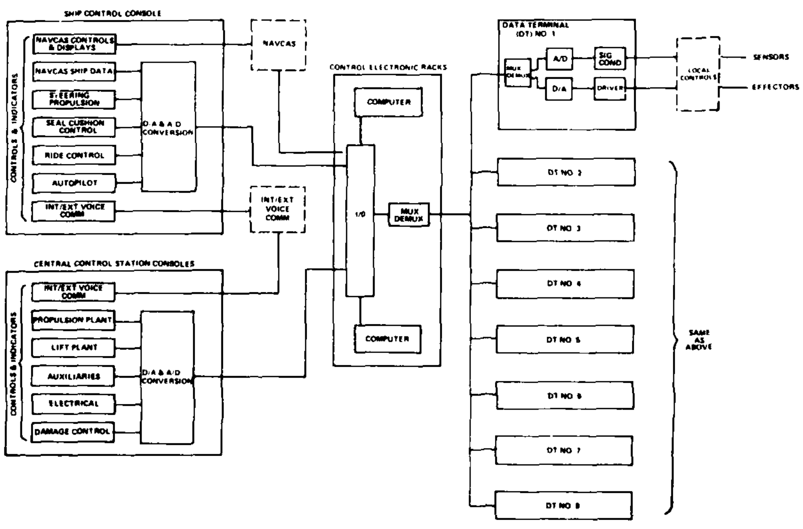


Figure 4. Functional block diagram of major ship control system elements.

data terminal) and conditioned and multiplexed prior to transmital to the Control Electronics rack where the various signals are relayed to the SCC and CCS. Although not shown, "hard wired" paths are provided for certain critical functions.

The SCC provides two virtually identical stations (Figures 5 and 6). The console is manned by a "Ship Control Officer" and an "Assistant Ship Control Officer." The "helm" normally resides with a single specific location; however, each station has full capability. Only the Propulsion and Lift Control Panels and the Ride Control and Autopilot Panels are shared. Although the navigation and collision avoidance system[1] (NAVCAS) is treated as a separate functional system, multi-purpose displays for collision avoidance data, surface search radar, and "electronic charts" are included at each station. The Commanding Officer's station is located just aft of the Propulsion Control Panel so that all controls settings and displays are readily visible.

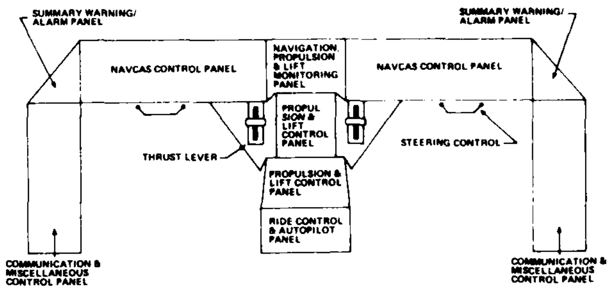


Figure 5. The Ship Control Console in plan view.

MANEUVERING

The 3KSES design, of which the above is probably most representative, will be unusual in that it is neither a prototype nor a pure technology ship; rather, it is intended to prove out the military utility of the SES concept. In addition, a goal of the SES design is to maintain manning at the minimum level consistent with the safety and operational requirements of an ocean going ship. The above design attempts to satisfy both these goals by normally operating with only two men at the CCS and with the previouly mentioned "Ship Control Officer" and "Assistant Ship Control Officer" at the SCC. As is apparent, it will be possible for the functional loading of the latter two operators to become quite high under certain conditions. The proposed solution to this problem is to supply a simple maneuver oriented control system for control of the ship in manual and emergency modes. This is intended to allow two operators to maintain control of the ship even in high loading situations. This same maneuver oriented system also includes a computer assisted primary mode whose function is to provide for automatic sequencing of effectors and yaw stabilization. This is intended to minimize operator loading during normal operation and fatigue over a normal watch period. The functional relationship between the elements of the Maneuvering Control System is indicated in Figure 7. Both a direct link and a computer assisted link is supplied to all ship effectors including the four engines, four waterjet steering sleeves, and the port and starboard thrust reversers.

Automatic sequencing of the effectors and the control logic is supplied by the Control Allocation Law which resides in the main control processor. The potential effectiveness of the Control Allocation Law is based upon explicitly incorporating all ship control effectors into the maneuvering control law. This is possible since a multiplicity of effectors can be used simultaneously to control the SES. This feature can be used to advantage in a number of ways of which the obvious choice is to use this redundancy to increase the "availability" of the ship. Since it is possible to maintain performance near the maximum practical capability of the ship by automatically sequencing out "failed effectors," this approach provides for a graceful degradation of ship performance without causing an excessive increase in complexity. In addition, the gas turbines which drive the waterjet thrusters can be very wasteful of fuel during low speed maneuvering. The control algorithm can be used to control fuel consumption during these maneuvers and, of equal or greater importance, to minimize engine cycling as well as cycling of other control effectors. A number of tradeoffs are required in order to establish the logic and algorithms for these control processes. It is of course necessary to assure ship saftey. This is acheived by means of operating procedures and careful programing of limit schedules. The other tradeoff is with regard to the sensitivity or priority of the various sub-algorithms: fuel conservation, effector ware, and quality of control. Priority is established by appropriate gain constants or schedules in the algorithms.

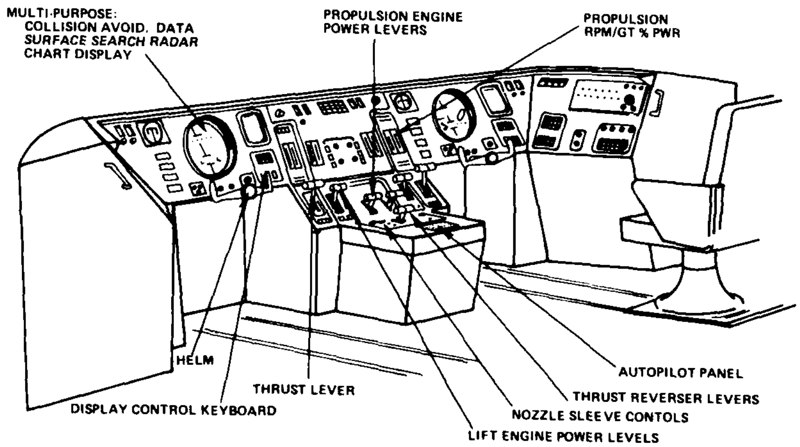


Figure 6. General arrangement of the Ship Control Station.

MANEUVERING ALGORITHM

Active control of the ship with respect to propulsion, steering, and reversing affects three degrees of freedom: surge, sway, and yaw. When the control axes are close to the principal inertial axes, ship motion about these axes is determined by forces and moments about the control axes with minimal cross-coupling. From the operators' point of view this is very desireable since the operator can easily identify the ship's response to his command. In particular, a separate propulsive sideforce control is of great advantage for low speed maneuvering since it can be used to facilitate docking operations, low sideslip turns, and general station keeping. At high speed a pure propulsive sideforce is not nearly so helpful since a combination of

propulsive turning moment and sideforce exists which can generate the desired turn rate for small sideslip angles. In fact, a particular turn can be acheived by a variety of thrust vectoring and thrust differential commands. This flexibility in implementing a desired maneuver, as will be discussed later, will be used to acheive additional ship capability.

Figures 8 and 9, which depict the maneuvering controls for the SES, divide the problem into two separate control processes: a high speed and a low speed maneuver. The high speed case includes all of the elements of the low speed case save for the sideforce command.

In the computer assisted mode three basic commands (ε_j, j = A, S, ω) are available to the ship operator (subscript A denotes forces and commands associated with forward and reverse thrust, subscript S denotes factors associated with sidethrust, subscript ω denotes factors associated with the turn command). These inputs from the helm and thrust level in turn must be scaled to the force and turn rate commands by a factor, F_j (j = A, S, ω), which accounts for the efficiency of that element as a function of speed (e.g.. the available thrust as a function of speed) and by the saturation, γ, of a given effector when presented with multi-axis commands (e.g., if a command input lies outside the operational envelope of the ship, the command signal will be reduced by a factor γ). In general it may not be possible to schedule all γ. In this instance some sensing procedure will be required.

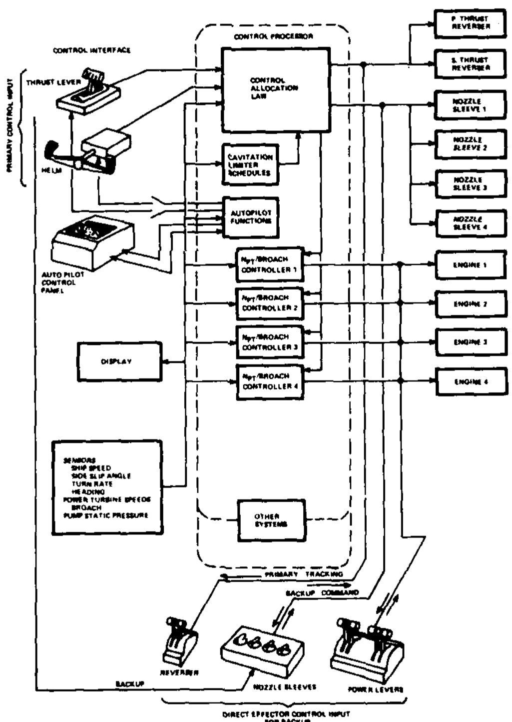


Figure 7. The maneuvering control system and ten effectors.

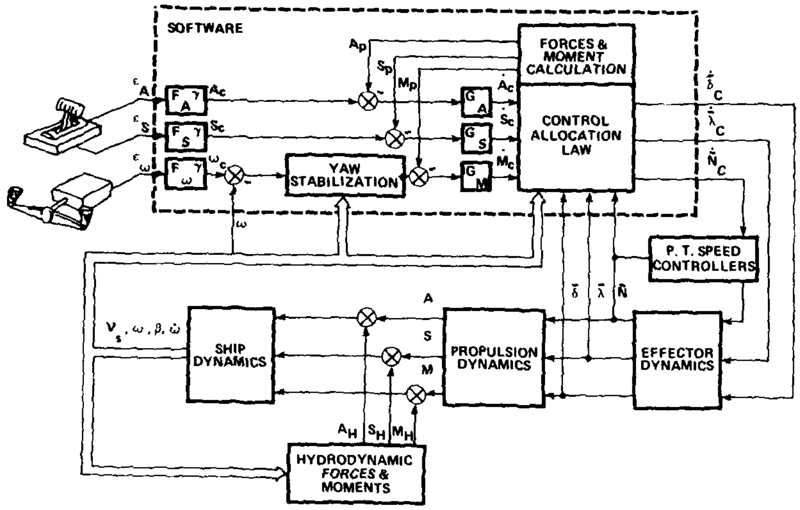


Figure 8. Functional elements of the low speed controller.

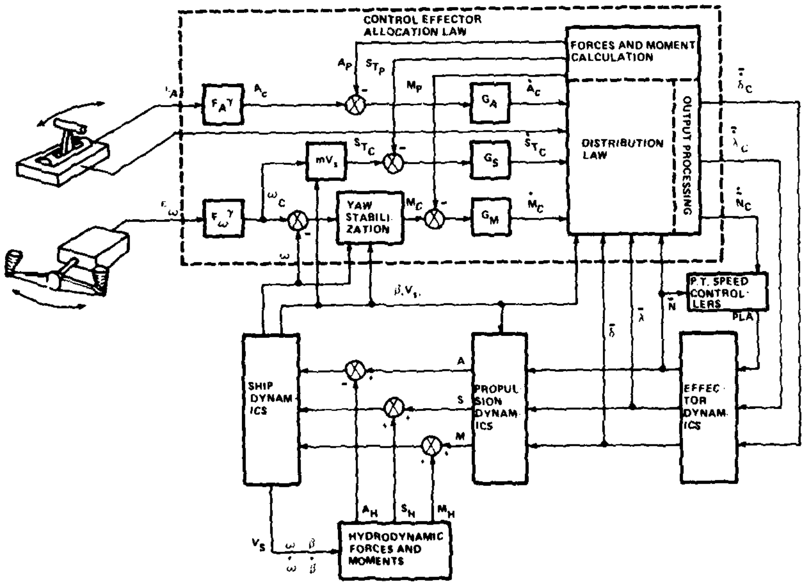


Figure 9. Functional elements of the high speed controller.

The maneuvering command values will thus be of the form:

$$j_C = \varepsilon_j F_j \gamma, \quad j = A,\ S,\ \omega \tag{1}$$

Now, for effective allocation of control effector commands, it is necessary that an adequate mathematical model of the contribution of the individual control effectors be available. Such a relationship has been established, for example, in Reference (2); however, these expressions are decidely non-linear. Linearization of these equations about a given operating point is not particulary attractive since realistc maneuvers are not in general characterized by small deviations from a reference state and since such a procedure would imply a great deal of scheduling and possible concurrent storage demands.

Fortunately, if the commanded forces and turning rate are written in terms of the control effector variables $\overline{N}$, $\overline{\delta}$, and $\overline{\lambda}$ (which describe engine speed, waterjet nozzle angle, and thrust reverser position):

$$j_C = j_C(\overline{N},\ \overline{\delta},\ \overline{\lambda}) \tag{2}$$

it becomes apparent that a linear relationship can be established between the rate of change of propulsive forces ($\dot{A}_C$, $\dot{S}_C$) and moment ($\dot{M}_C$), and the control variable rates ($\dot{\overline{N}}$, $\dot{\overline{\delta}}$, $\dot{\overline{\lambda}}$) simply by differentiating equation (2).

This relationship is still not in a particularly attractive form since it implies that the command input must be the time derivative of the command forces and moments. Fortunately again, this result can be acheived either by direct differentiation or by feedback techniques. The latter is of course to be preferred since the control error can be maintained small by comparing the command input to the desired output. This will in turn enhance the accuracy of those equations used in the mathematical model. The control inputs for the distribution law used in the low speed case (Figure 8) are thus:

$$\dot{j}_C = G_j\ (j_C - j_P), \quad j = A,\ S,\ M \tag{3}$$

where: G_j = the control gains

j_P = predicted propulsive forces and moment as derived from physical modeling

j_C = commanded propulsive forces and moment.

As indicated in Figure 9, in the high speed case (above about 10 knots), the derivative of the sideforce command, $\dot{S}_c$, is not used. Instead, a derived expression is used for the total calculated sideforce S_{TP}:

$$S_{TP} = mV_S\ (1 + \dot{\beta}) \tag{4}$$

where: m = ship total mass

V_S = ship speed

ω = yaw rate

$\dot{\beta}$ = sideslip angle rate.

It follows that for small sideslip angles, the total radial turning force is essentially the sum of the hydrodynamic sideforce and the propulsive sideforce. A particular rate of turning can therefore be acheived by some combination of these two components, i.e., some combination of thrust vectoring and thrust differential. The longitudinal force command is introduced by thrust level deflection while the lateral motion command, in the form of a turn rate command, is introduced by deflection of the

helm. The latter signal is the input for the yaw stabilization loop[2] indicated in Figure 10. The loop provides yaw stabilization by generating a command which is proportional to the yaw rate and the deviation from the yaw rate command:

$$M_c = - K_2 S\omega + (K_1 + \frac{K_0}{S}) (\omega_c - \omega) \tag{5}$$

where: $\omega_C = \varepsilon_\omega F_\omega \gamma$.

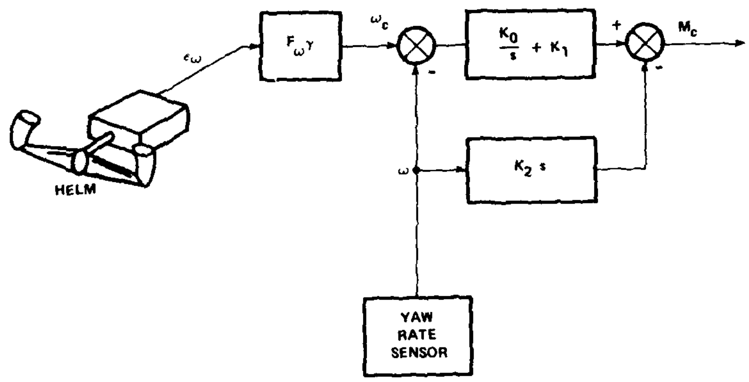


Figure 10. *The Yaw Stabilization Controller.*

Formulation of the Optimal Controller

The preceding dicussion has developed the relationship between the operator input commands, A_C, S_C, ω, and the available control effectors, or more precisely, between the rate commands and the control effector command variables $\dot N_i$, $\dot\delta_i$, $\dot\lambda_i$. For the low speed case these relationships can be summarized by:

$$\begin{bmatrix} \dot A_c \\ \dot S_c \\ \dot M_c \end{bmatrix} = \bar S \begin{bmatrix} \dot N \\ \dot \delta \\ \dot \lambda \end{bmatrix} \tag{6}$$

where: $\bar S$ = a 3X10 matrix of coefficients which are a function of N_i, δ_i, and λ_i

$$\dot N = \begin{bmatrix} \dot N_1 \\ \dot N_2 \\ \dot N_3 \\ \dot N_4 \end{bmatrix} \qquad \dot\delta = \begin{bmatrix} \dot\delta_1 \\ \dot\delta_2 \\ \dot\delta_3 \\ \dot\delta_4 \end{bmatrix} \qquad \dot\lambda = \begin{bmatrix} \dot\lambda_1 \\ \dot\lambda_4 \end{bmatrix}$$

The values of the command rates, $\dot{j}_C$ (j = A, S, M), are obtained by comparing the input commands, j_C, to predicted forces and moments, j_P, which have been calculated from the detailed ship model. Expression (6) thus represents a system of three equations in the ten effector commands $\dot{N}$, $\dot{\delta}$, $\dot{\lambda}$. Activation of either reverser requires the centering of the corresponding waterjet nozzle so that only eight independent commands are available.

Mathematically we have a system of three equations with eight degrees of freedom. This redundancy can be used to great advantage to limit effector activity and fuel consumption, and to increase the availability of the ship control system. The former is achieved by using a cost function[3] of the form:

$$g = K_f \sum_{i=1}^{4} \dot{N}_i N_i^{5/4} + K_n \sum_{i=1}^{4} \dot{N}_i^2 + K_\delta \sum_{i=1}^{4} \dot{\delta}_i^2 + K_\lambda \sum_{i=1,4} \dot{\lambda}_i^2 \tag{8}$$

The various K_i are weighting factors which determine the relative importance of the activity of the different classes of effectors. The first term (proportional to $N^{5/4}$) is a derived cost function for the fuel consumption of a gas turbine. The next three terms minimize activity of the engines, waterjet nozzles, and reversers by penalizing large values of the effector rate commands. It should be noted that such a scheme could be adapted to allow for preferred modes of operation. As an example, K_f could be set equal to zero for high speed operation to assure minimum engine cycling during normal operation, but could be set equal to one for low speed operation to minimize fuel consumption during docking or station keeping.

Increased availability is achieved by means of a sequencing system which allows the ship to maneuver at or near peak performance despite the loss of one or more control effectors. One method of realizing this goal is by incorporating a series of effector-enable-flags which are either 0 or 1 depending or whether the corresponding effector is disabled (due to either failure or specific mode selection) or enabled.

Formally if we introduce scaled variables, $\dot{n}_i$, $\dot{\zeta}_i$, and $\dot{\xi}_i$ and matrix H, x, c, and p defined by:

$$\dot{n}_i = K_n^{1/2} \dot{N}_i, \qquad \dot{\zeta}_i = K_\delta^{1/2} \dot{\delta}_i, \qquad \dot{\xi}_i = K_\lambda^{1/2} \dot{\lambda}_i$$

$$\begin{aligned} H_{ij} &= K_n^{-1/2} f_j, && j = 1, 2, 3, 4 \\ &= K_\delta^{-1/2} f_j, && j = 5, 6, 7, 8 \\ &= K_\lambda^{-1/2} f_j, && j = 9, 10 \\ &= 0, && i = j \end{aligned} \tag{9}$$

$$x = \begin{bmatrix} \dot{n} \\ \dot{\zeta} \\ \dot{\xi} \end{bmatrix}, \qquad c = \begin{bmatrix} \dot{A}_c \\ \dot{S}_c \\ \dot{M}_c \end{bmatrix}$$

$$p = \frac{K_n^{-1/2}}{2} \left[N_1^{5/4} f_1, N_2^{5/4} f_2, N_3^{5/4} f_3, N_4^{5/4} f_4, 0, 0, 0, 0, 0, 0 \right] \tag{10}$$

equation (6) becomes:

$$c = \bar{S} H x \tag{11}$$

and the cost function g becomes:

$$g = 2p^T x + x^T x \tag{12}$$

where T denotes the transpose of the respective matrix. The controller thus minimizes the cost function, g, by solving for the optimal set of effector commands, x, and includes those constraints necessary to account for failed effectors.

Although somewhat complex in description, the Control Allocation Law thus supplies a reasonably simple algorithm for sequencing the multiple effectors available for ship control. The major benefits of this approach are decreased operator loading and enhanced available for the ship maneuvering system. An additional feature is the capability to adjust the activity of the individual effectors based on field experience.

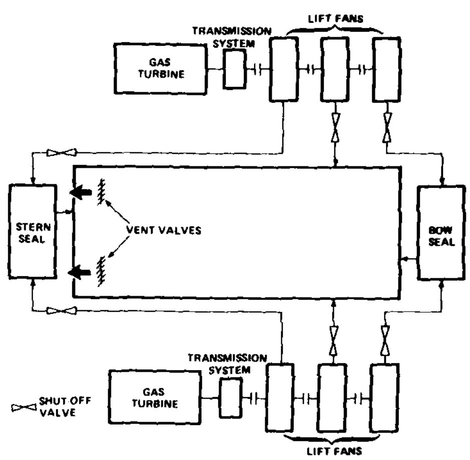


Figure 11. The variable geometry fans are an integral part of the lift fans. The vent valves are connected directly to the main plenum.

RIDE CONTROL SYSTEM (RCS)

The other major task of the ship control system is to provide attenuation of the heave motions of the SES. This is accomplished by the RCS which controls pressure in the SES plenum and which consequently controls the heave motions of the SES. Figure 1 indicates the general physical relationship of the lift system elements and seals; while Figure 11 provides a schematic of the lift system air flow.

Lift system fans are driven by gas turbines through a series of reduction gears. The fans are centrifugal flow, double axial inlet fans with an inlet variable geometry (VG) feature. All lift fans are identical but differ in actual installation orientation. The air distribution system transfers air from the fans to the seals and main cushion by means of independent duct systems. The forward fans discharge air directly into the bow seal via a short diffuser while the center fans have a similar discharge path to the main cushion. The flow of the aft-most fans is transmitted to

the stern seal by a slightly longer combination of round and square ducts followed by another set of short diffusers. The lift system effectors which can be controlled by the RCS are VG fan actuators, vent valve (VV) actuators, and the rotational speed of the VG fans. The RCS establishes the operating point of the system, i.e., establishes the reference or bias settings of the actuators and fan speed, and, provides feedback as necessary to regulate cushion pressure and to control the crafts vertical plane motions (heave, and potentially pitch and roll) based upon the sensing of either heave acceleration or plenum pressure.

The VG fans, as the name implies, are fans whose general configuration or geometry can be varied in order to extend the useful operating range of the fans. In the case of radial fans, the angle of attack of the blades is typically varied, while in the case of centrifugal fans, the effective vane area is varied. The variable geometry feature of the centrifugal fans consists of two translating sleeves, one at each inlet, mounted concentrically around the axis of rotation of the fans. Movement of the sleeves along this axis effectively shrouds the inlet and varies the effective inlet area. The sleeves can be positioned at a fixed point or modulated continuously. The steady state feature is utilized to provide adequate stall margin for steady state operation, while the modulation capability is used during dynamic ride control. Total travel capability of the sleeves in the latter mode is about 17 inches with a half power point corresponding to about four Hertz. Typical cushion pressure for a 3000 ton SES will range from 200 to 330 PSF depending on operating conditions. Nominal corresponding air flow will vary from about 32,000 to 38,000 CFS for a 40 to 90 knot range of ship speed, with peak flows to about 62,000 CFS. The VV are basically openings or apertures in the deck. Their area is modulated by rotating vanes or shutters mounted in the deck openings. Typical cushion pressure for a 2000 ton SES would range from approximately 180 to 290 PSF. The nominal corresponding vent valve flow would range from near zero to a maximum of 40,000 CFS at an effective venting area of 144 square feet.

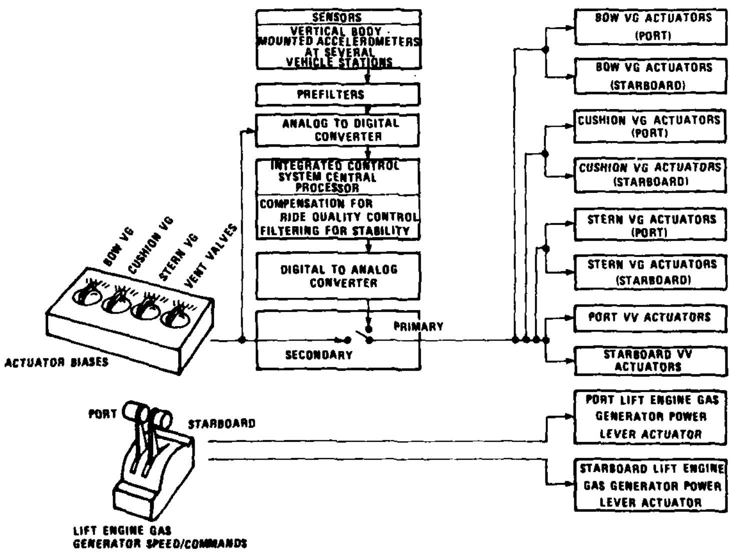


Figure 12. Lift and Ride Controller functional block diagram.

The RCS can operate with either VG fans or VV or both. The configuration now considered to have the best performance consists of VG fans and VV operating in a push-pull mode. The VG actuators are biased partially "open" and serve to provide the principal control. The VV actuators are biased "shut" and only open when the venting requirements exceed a specified level. This configuration[4] has been shown to be the most efficient in terms of fuel consumption. The VG fans are basically more fuel efficient than the VV; but the inclusion of the VV in this configuration provides a wider range of control.

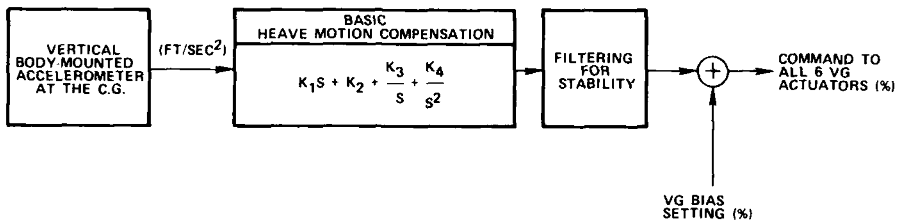


Figure 13. Block diagram of a Ride Controller using acceleration feedback.

Stabilization

Figure 12 depicts the functional relationship between the elements of the lift and ride control system. A signal derived from either the ship heave acceleration or cushion pressure is used to provide feedback for the Ride Controller. The sensed signal is conditioned and fed to a controller which consists of a heave mode compensation network and a stabilization filter (see Figure 13 where for simplicity only the VG fan control is indicated). A variety of networks[5] have been studied for application in the stability filter. The basic choices are (1) filtering via vector cancellation, (2) filtering via sensor location, (3) linear filtering, and (4) nonlinear filtering. Filtering via vector cancellation involves separate filtering of each transducer signal before averaging or summing of the total signal. This approach follows from the fact that for frequencies at or below the basic heave mode frequency region, pressure components respond in phase in both the bow and stern seals and throughout the plenum. At higher frequencies, phase differences develop between the various pressure elements. It is thus possible in principal to phase the signals from two or more transducers in such a manner as to cancel each other in specific frequency regions. This method would allow filters to be rolled off at higher frequencies than would be the case with linear filtering alone, thus allowing less phase lag and consequently providing better control. In practice this approach has not been widely applied.

From the previous dicussion, it is evident that the phase and magnitude of the transducer signal are location sensitive and thus provide the potential for stability control. Again this approach has had little practical application. Application of linear filters to the stabilization problem are reasonably straight forward except in those instances when structural or pneumatic modes frequencies fall in the close proximity to the basic heave resonance of the ship. When such an instance occurs a very elaborate linear (higher order) filter may be required to separate the modes.

Still another approach is the use of a non-linear filter of which the Independent-Magnitude-Phase (IMP) filter[6] is one example. The IMP filter has been considered because it is capable of providing both independent magnitude and phase as

indicated in Figure 14. The signal is first separated into two channels. In the first channel the signal passes through a linear filter, $F_1(s)$, whose output is then passed through a bistable element. In the second channel the signal is passed through another linear filter, $F_2(s)$ and the absolute value of the filtered signal is then taken. Finally the outputs of the separate channels are recombined by multiplication. As configured in Figure 14, networks $F_1(s)$ provides phase lead while $F_2(s)$ provides signal attenuation. The typical resultant transfer function is indicated in Figure 15. The filter obviously achieves the desired result of good high frequency isolation with minimal lag; of course, careful attention to real world considerations such as noise background and quantization effects is required.

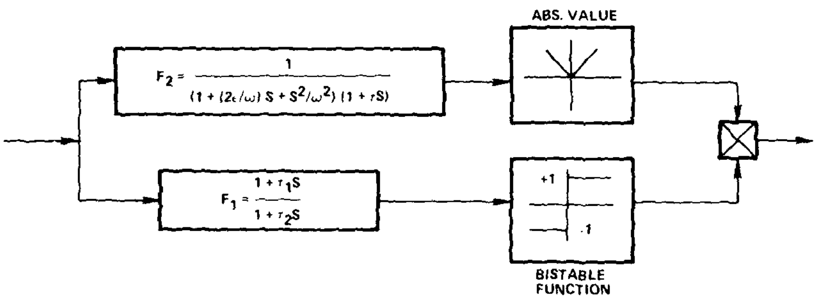


Figure 14. Block diagram of an Independent Magnitude and Phase (IMP) filter.

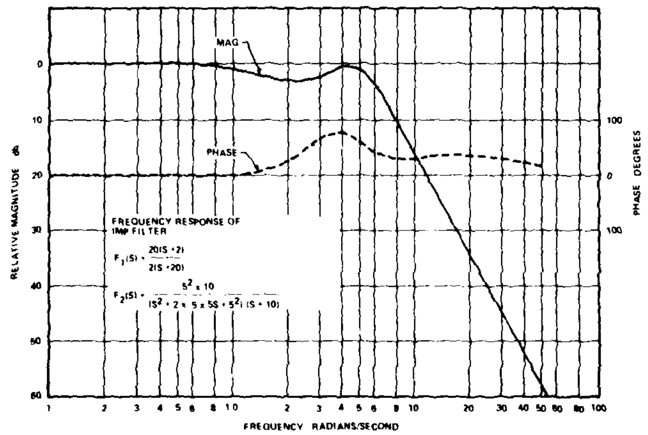


Figure 15. Frequency response of a typical IMP filter.

Heave Mode Compensation

Early in the development of the SES, a mathematical model[7] for predicting the loads and motions of the SES was established. Motion data from this modeling was in turn used in man-machine simulations[8] which established the desirability of a RCS for SES operation in high sea states. Work (see Reference (9) for example) has continued on both the modeling and the RCS design since that time so that both five Degree-of-Freedom (DOF) frequency domain and six DOF time domain simulations are currently available. The resulting equations of motion, even when linearized[10] are still rather formidable. In practical application the design of the RCS has been primarily restricted to an initial simulation of two DOF, pitch and heave, followed by a more elaborate checkout simulations. (This has been true for two reasons: (1) current designs only attempt to deal with heave motion, (2) the preponderance of motion is associated with the heave and pitch DOF.) Lags for control actuators and air inertia are incorporated as are non-linear describing functions to account for saturation of the flow capability of the vent valves and the VC fans. The various gains are then optimized in terms of this limited simulation and are finally tested for overall effectiveness using the more accurate 6 DOF simulations. Despite its seeming unwieldiness, such a paramaterization allows a reasonably rapid approach to practical gain values.

For the purpose of discussion, it is still possible to consider only the two DOF modeling. Based on this representation and including only first order effects in the frequency region of the basic heave motion, the transfer function for the acceleration of the center of gravity, a_{cg}, relative to the wave pumping of the cushion volume, $\dot{V}_W$, is given in terms of the heave mode natural frequency, ω_H, the heave mode lag frequency, ω_L, and the heave mode static gain, K_0, by[4]:

$$\frac{a_{cg}}{\dot{V}_W} = \frac{K_0 S^2}{\left(1 + \frac{S}{\omega_L}\right)\left(1 + \frac{2\zeta_H S}{\omega_H} + \frac{S^2}{\omega_H^2}\right)} \tag{13}$$

$$\omega_L = \frac{\partial Q}{\partial P} \tag{14}$$

$$\zeta_H = \frac{1}{2}\left(\frac{Q}{2\Delta P_C} - \frac{\partial Q}{\partial P}\right)\left(\frac{m\gamma}{A_C^3}\frac{P_T}{h_C}\right)^{\frac{1}{2}} \tag{15}$$

$$K_0 = \left(\frac{\partial Q}{\partial Z}\right)^{-1} \tag{16}$$

where:

ΔP_C = cushion gage pressure (PSF)

P_T = cushion pressure (PSF)

Q = total system flow rate (CFS)

$\left(\frac{\partial Q}{\partial P}\right)$ = fan slope (CFS/PSF)

$\left(\frac{\partial Q}{\partial Z}\right)$ = leakage as a function of vehicle and seal motion (CFS/PSF)

A_C = cushion area (ft^2)

m = total vehicle mass (slugs)

γ = specific heat of air

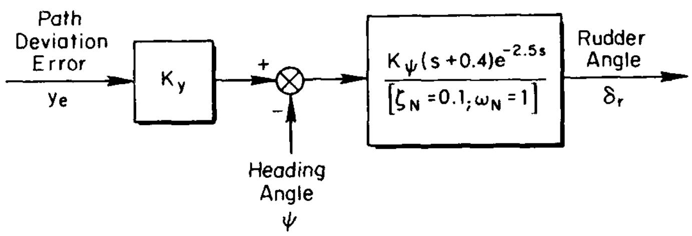


Figure 6. Summary Block Diagram of Helmsman's Control Response.

CONCLUSIONS

An examination of the open-loop steering describing function measurements in the SES simulation has shown that, regardless of sea motion condition, both helmsmen adopted average heading and displacement gains so as to maintain the closed-loop displacement bandwidth in the neighborhood of 0.2 rad/sec with acceptable margins of stability in phase and gain. This is an adequate bandwidth for course-keeping at SES cruising speeds. Additional adopted first-order lead equalization appears within the heading loop between 0.3 and 0.5 rad/sec to maintain a well-damped characteristic directional mode of the ship. This lead equalization may be generated by perceiving the turn rate indicator.

Although no evidence for biodynamic amplification appears in these results, relatively lightly damped and amplified steering wheel dynamics appear at about 1.25 rad/sec as an intended artifact of the simulation. The relatively low damping is due in part to the lower-than-recommended values of Coulomb friction provided in the simulated artificial feel for the steering wheel to reduce breakout torque. However, the spring gradient in the steering wheel was also too low. Consequently, the natural frequency of neuromuscular actuation was too low, and the helmsman reduced the inherent steering wheel damping ratio even more, because a portion of his time delay exists within his perceived rudder angle feedback loop.

That the helmsmen were scanning the rudder angle indicator is corroborated by their commentary. This is presumably a part of the transfer of training for steering low speed displacement-hull ships. However, with a stiffer and more linearly damped artificial feel system with a lower breakout force, the proprioceptive feedback available from the steering wheel will provide a superior equivalent to visual rudder angle feedback from the panel instrument. As a result, the neuromuscular dynamics will be desirably higher in frequency and suppressed in amplitude. This will yield a better margin of stability and improved course-following performance in aggravated seas.

The overall effective time delay of the helmsmen estimated from the open-loop describing function measurements is between 2 and 3 sec. This is about 2 sec larger than would be expected to accompany the adopted equalization when using the electronic horizontal situation display. Such a relatively large time delay may be caused by scanning delays among the helmsman's instruments and visual field and by a division of attention among his other tasks during the measurement interval. We therefore recommend training in the more effective use of integrated horizontal situation displays for steering.

c) ζ_N is either 0.08 (Crew A) or 0.15 (Crew B) to fit the describing function amplitude at the third DFA measurement frequency. These relatively low values of ζ_N are due in part to the lower-than-recommended values of Coulomb friction provided in the simulated artificial feel for the steering wheel. However, they also represent the fact that the helmsman will reduce the inherent steering wheel damping ratio anyway, because a portion of his time delay exists within his perceived rudder angle feedback loop or his proprioceptive steering angle feedback loop.

d) τ_d is between 2 and 3 sec to fit the describing function phase angle at the third DFA measurement frequency.

e) Y_y is a pure gain and the gain product $Y_y K_\psi$ is adjusted so that the amplitude of M/E matches that at the lowest DFA measurement frequency.

The quite satisfactory results of the fitting procedure for the helmsman in Crew B are displayed in Fig. 4 and tabulated in Table 2 for both helmsmen, and a summary block diagram equivalent is shown in Fig. 6.

Table 2. Numerical Values Representing the Helmsmen.

	Crew A	Crew B
Y_ψ/Y_y (ft)	791.	791.
$Y_y K_\psi$ (rad/ft)	0.0000927	0.0000981
$Y_\psi K_\psi$	0.0733	0.0776
ω_N (rad/sec)	1.25	1.25
T_L (sec)	3.	2.5
ζ_N	0.08	0.15
τ_d (sec)	2.9	2.4
ω_c (rad/sec)	0.18	0.19
Phase margin (deg)	29.	28.
ω_u (rad/sec)	0.5	0.525
Gain margin (dB)	7.	8.

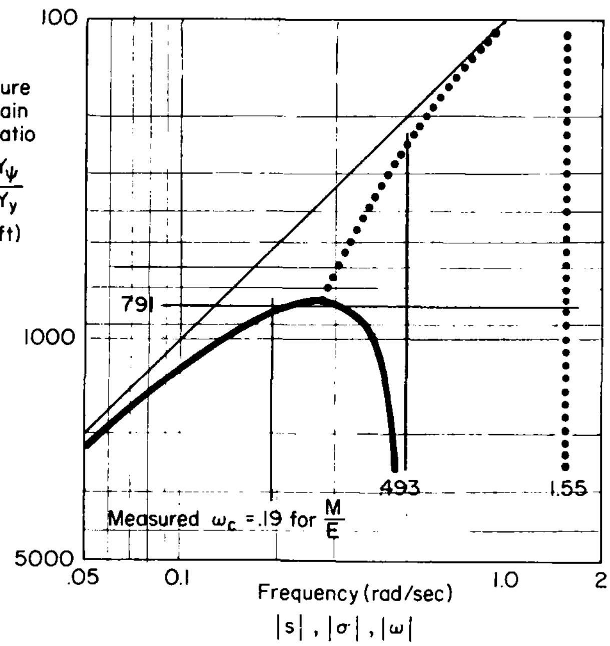


Figure 5. Bode Root Locus of the Zeros of $[Y_\psi s N^r_{\delta_r} + Y N^{\ddot{y}}_{\delta_r}]$ as a Function of the Pure Gain Ratio Y_ψ/Y_y for $|Y_\psi| >> |Y_y|$ (Refer to the Numerator of Eq. 3)

$$F_\psi = \frac{K_\psi(1 + T_L s)e^{-\tau_d s}}{\left[1 + \frac{2\zeta_N s}{\omega_N} + \frac{s^2}{\omega_N^2}\right]} \tag{5}$$

Numerical estimates for the parameters in $F_\psi Y_y$ were based on the following observations and conditions:

a) T_L is approximately 2 or 3 sec to provide a phase crossover frequency in the neighborhood of the second DFA measurement frequency, 0.5 rad/sec, with the 5 or 6 dB gain margin measured and to maintain a well-damped nonoscillatory characteristic directional mode of the ship emanating from Δ.

b) ω_N is slightly in excess of 1 rad/sec and approximately equal to the third DFA measurement frequency, 1.25 rad/sec.

Substitution in Eq. 3 of the typical numerical values for the ship's lateral-directional transfer functions in Table 1 will reveal the following key points:

- M/E is of the form K/s^2 at low frequency as we would expect.
- $|Y_\psi| >> |Y_y|$ in order that $sN^r_{\delta_r}$ may provide the requisite low-frequency lead equalization to convert M/E to the form K/s in the region of unit gain crossover.
- The characteristic directional oscillatory mode of the ship is quite low in frequency (approximately 0.5 rad/sec) but well damped.
- Y_ψ (or F_ψ) should adopt lead equalization in the vicinity of 0.5 rad/sec to maintain a well damped closed-loop directional mode.
- The characteristic rolling oscillatory mode is higher in frequency (approximately 1.5 rad/sec), lightly damped, but suppressed in amplitude by the zeros of the lateral and directional numerators.

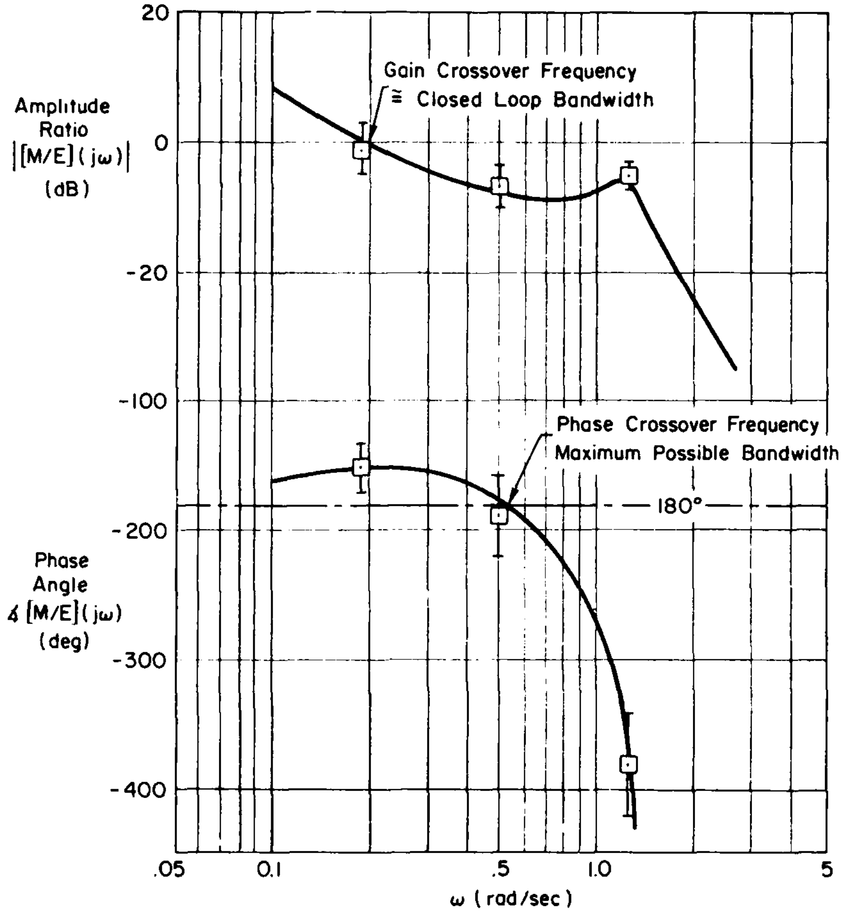
We shall next discuss the procedure employed to establish F_ψ, Y_ψ, and Y_y which will provide plausible heading and lateral displacement control techniques to interpret the response measured by the DFA. We shall illustrate the procedure by adopting pure gain equalization within Y_ψ and deferring the mid-frequency lead equalization to F_ψ, although equivalent results can be obtained by adopting the converse equalization technique because $|Y_\psi| >> |Y_y|$. In either case pure gain equalization will suffice for Y_y.

It was first necessary to determine the behavior of the numerator zeros of Eq. 3 as a function of Y_ψ/Y_y and to select an appropriately large value for the gain ratio Y_ψ/Y_y which would fix the location of the zeros for the overall lateral displacement control so as to provide the low-frequency lead equalization apparent in the measured response. The Bode root locus of the numerator zeros was therefore generated as depicted in Fig. 5. The gain ratio of 791 was selected for Y_ψ/Y_y to provide M/E with a reasonable frequency interval having the form approaching K/s in the neighborhood of the unit gain crossover frequency, 0.19 rad/sec estimated from the DFA.

With the gain ratio of Y_ψ/Y_y thus established, M/E is now represented partially in numerical form in Eq. 4, by allowing Y_ψ to covary with Y_y in accord with the constant ratio selected above.

$$\frac{M}{E} = \frac{739\ Y_y(0.213)(0.315)[0.192,1.556]}{s^2\Delta} F_\psi \qquad (4)$$

It now remains to determine the product $F_\psi Y_y$ so that M/E in Eq. 4 will fit the set of three measurements in Fig. 4. Realizing that the helm itself was a relatively lightly damped spring-restrained steering wheel with a very low undamped natural frequency ω_N on the order of 1 rad/sec, we shall hypothesize that the important features of F_ψ can be represented by:



Measurements represent the ensemble average and plus or minus one standard deviation of six runs by the helmsman in Crew B. The theoretical fitted describing function is based on Eq. 3 and represents the combination of the SES, the helmsman, displays, and steering control.

Figure 4. Example of Open-Loop Describing Function Measurements at the Rudder Control Point for Manually Controlled Course-Keeping with a Simulated 2000T SES.

loop lateral displacement control technique from rudder error to helmsman's output is obtained from inspection of Fig. 1.

$$\frac{M}{E} = \frac{Y_\psi s N^r_{\delta_r} + Y_y N^{\ddot{y}}_{\delta_r}}{s^2 \Delta} F_\psi \qquad (3)$$

An attempt was made to determine the influence, if any, of performing the subcritical speed control task on the measurements of M/E for the steering tasks. No influence is evident in the two runs for the helmsman in Crew B. However, for the helmsman in Crew A only one run with both tasks is available; its measurements appear more noisy with evidence of increased time delay in the steering task, and the phase crossover frequency apparently decreases from a value in excess of 0.5 rad/sec to about 0.2 rad/sec as a result of the division of attention. Yet, there is no evidence of a corresponding gain reduction by the helmsman in Crew A to provide more than 2.5 dB gain margin of stability when both tasks are being performed. There also was the possibility of biodynamic coupling from heaving motions to steering motions when the helmsman had only his right hand on the wheel while performing the speed control task with his left hand. Although we shall identify neuromuscular dynamic amplification in the results at the third measurement frequency, there is no evidence for biodynamic causation.

The mean values and standard deviation of M/E for six selected runs by the helmsman in Crew B are presented in Fig. 4. Four are runs for which the helmsman performed only the steering task, and two are runs for which he performed both the steering and speed control tasks simultaneously. The amplitudes and phase angles for M/E are presented only at the three lowest measurement frequencies corresponding to 0.1884, 0.5014, and 1.256 rad/sec, because M/E at the two higher measurement frequencies had very low signal-to-noise ratios and was dominated by noise. The average unit gain crossover frequency of the lateral displacement loop closure by each helmsman is slightly below 0.2 rad/sec, and the average phase crossover frequency is in the vicinity of 0.5 rad/sec.

The excessively variable and noisy measurements obtained here are, in part, the result of deliberately violating a caveat in using the DFA, viz., maintaining input magnitude sufficient to yield reasonable displayed error deviations(8). This caveat was regretfully sacrificed in favor of crew motivation, because the helmsmen complained that higher input amplitudes produced abnormally great activity in turn rate and rudder displacement based on their experience, even though the input disturbance was applied to the innermost loop available at the rudder control point and not directly displayed on the rudder angle indicator. This resulted in a K/ω^2 power spectrum in displayed turn rate, whereas one should in future tests employ a sum of approximately equal amplitude sinusoids for the input at the rudder. This will allow the SES controlled element to shape the power spectra of the displayed signals and will yield better signal-to-noise ratios in the measurements. In retrospect, we placed too much weight on crew experience, because at that point in time neither helmsman had had experience steering an SES in aggravated seas at cruising speeds by reference to instruments.

We shall now describe the rationale for partitioning M/E between the (known) controlled element and the helmsman in such a way as to infer forms for his describing functions. The results of applying these inferred forms within a theoretical model for M/E and fitting the same to the measurements are plotted in Fig. 4.

MANUAL CLOSURE OF HEADING AND LATERAL DISPLACEMENT LOOPS LEADING TO AN EXPLANATION OF THE MEASURED DESCRIBING FUNCTIONS

The topology of the helmsman's control technique represented in Fig. 1 is founded on foreknowledge of the K/s^2 form of the SES controlled element and includes the necessary heading and lateral displacement loops with provision for lead equalization in the heading loop. The helmsman is represented by partitioned describing functions F_ψ, Y_ψ, and Y_y. The describing function, M/E, of the open

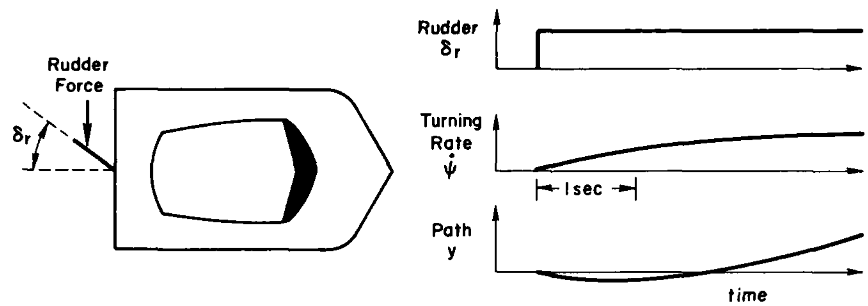


Figure 3. Exemplary Time Histories of Surface Effect Ship Dynamic Responses to a Step Change in Rudder Angle.

The lags and non-minimum phase characteristics combine to cause apparent ship response delays in lateral displacement on the order of several seconds, which makes for a challenging dynamic control task. The various displays available to the helmsman help by allowing him to control intermediate states or derivatives of the final lateral displacement. In Fig. 1 the Y_y, $Y_{\dot{\psi}}$, and F_{ψ} terms account for the helmsman's operation on the variety of displayed information.

DESCRIBING FUNCTION MEASUREMENTS WITH MANUAL CONTROL

The forcing function provided by the DFA is labeled "I" in Fig. 1, where it is injected into the common path of the control loops as a disturbance by summation with δ_r, the helm's steering signal to the rudder. If the disturbance I is viewed as a "command" input and the helmsman's rudder signal $\delta_r \equiv -M$, the negative motion feedback, the rudder error $\delta_e = E(= I - M)$ acts to disturb heading and to displace the ship laterally from the desired course. Thus, for our purpose, E represents an acceptable measure of closed-loop performance, the finite Fourier transform of which is computed by the DFA, and $[E/I](j\omega)$ represents the closed-loop error-to-input describing function.

The open-loop describing function of the helmsman's lateral displacement control technique in combination with the known mathematical model of the SES can be represented by $[M/E](j\omega)$, which may be obtained from the relationship:

$$[M/E](j\omega) = \frac{1 - [E/I](j\omega)}{[E/I](j\omega)} \tag{2}$$

After identifying that portion of $[M/E](j\omega)$ which represents the known SES dynamics, the remaining portion will represent the combined equivalent of the loops, gains, frequency-dependent equalization, and steering dynamics involved in the helmsman's control technique. A digital computer program based on Allen(5) is used for identifying $[M/E](j\omega)$ and partitioning the result between the machine and the man.

Some of the results of applying that program to the SES simulation have been analyzed. We have found no significant differences in M/E which correlate with the different sea motion conditions tested. The only significant difference between helmsmen is in the average amplitude of M/E at the third measurement frequency, which is about 6 dB higher for the helmsman in Crew A than in Crew B.

Table 1. Typical Lateral-Directional Open-Loop Control Response Transfer Functions for a 2000T SES at Cruising Speed

Denominator:

$$\Delta(s) = [s^2 + 2(0.898)(0.469)s + (0.469)^2] \times [s^2 + 2(0.198)(1.55)s + (1.55)^2]$$

$$\Delta(0) = 0.528 \text{ sec}^{-4}$$

Numerator for yaw rate response to rudder angle:

$$N^{r}_{\delta_r}(s) = 0.999(s + 0.493)[s^2 + 2(0.191)(1.55)s + (1.55)^2]$$

$$N^{r}_{\delta_r}(0) = 1.185 \text{ sec}^{-5}$$

Numerator for lateral acceleration response to rudder angle:

$$N^{\ddot{y}}_{\delta_r}(s) = -52.0(s + 0.983)(s - 1.03) \times [s^2 + 2(0.191)(1.51)s + (1.51)^2]$$

$$N^{\ddot{y}}_{\delta_r}(0) = 120 \text{ ft/sec}^6\text{-rad}$$

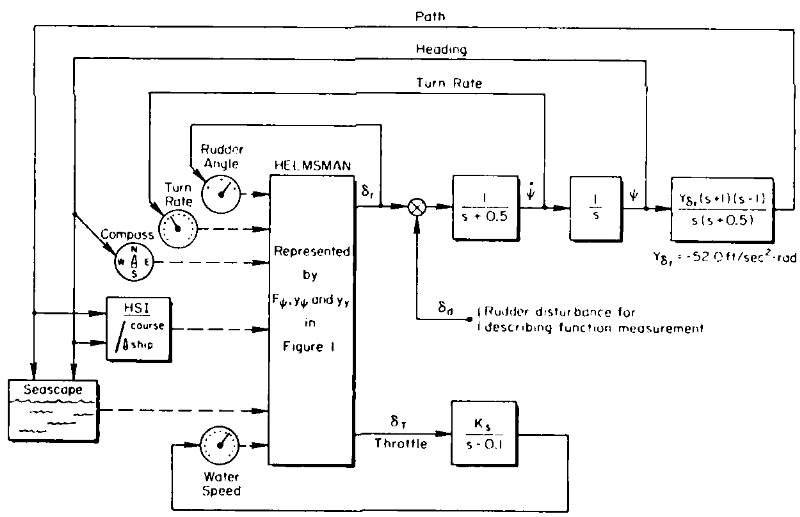


Figure 2. Helm Control Tasks.

multiloop response properties with a single disturbance input as used in these experiments are described in Teper[6] and McRuer[7] where the principles are applied to pilot control of hovering vehicles and driver control of highway vehicles. The SES with its broad beam and side wall keels exhibits little rolling and sideslipping in common with catamarans and highway vehicles.

The helm itself was a 14 in. diameter marine steering wheel. An artificial feel spring was provided, and the wheel breakout torque and the kinetic Coulomb friction damping torque were adjusted to be less than those values currently used in order to examine more critically the possibilities for neuromuscular coupling, biodynamic amplification thereof, and closed-loop steering stability limitations. The steering wheel rotation from stop to stop was 270 deg and the corresponding rudder travel, 30 deg, thereby giving a 9:1 steering ratio as in contemporary practice.

The helmsman was asked to perform the steering-only task with both hands on the wheel to minimize the possibility of biodynamic coupling. In addition to the primary steering control task, a sub-critical* speed regulation task provided a secondary surrogate for trimming the water speed of the craft. Speed was regulated with the helmsman's left hand on a friction-restrained quadrant throttle lever with 60 deg of fore-aft travel, so that when the helmsman was instructed to perform both steering and speed control tasks, he maintained only his right hand on the steering wheel.

The ship's lateral-directional dynamic motions in response to the helm and disturbance were represented by linear constant coefficient differential perturbation equations in body axes with respect to a trimmed condition at constant speed. The ship's cab was assumed to be located at the center of gravity in developing the lateral-directional transfer functions given in Table 1 for a trimmed cruising condition.

The differential equation representing the divergent (unstable) controlled element for speed regulation, which was independent of the equations of lateral-directional craft motion, is given by Eq. 1:

$$\dot{u} = \lambda u + K_s \delta_t \qquad (1)$$

The inverse time constant λ was chosen as 0.1 rad/sec for the simulation to approximate a slow sub-critical divergence which would require consistent but not overwhelming attention to the side task of speed regulation. Comments by the helmsman attested that the speed regulation task and the steering task together saturated the helmsman's workload. Yet there was only one out of four runs with both tasks where performance on the steering task appeared to degrade.

The helmsman's tasks are summarized in Fig. 2. Approximations to the lateral dynamics of Table 1 are shown to illustrate the low-frequency (i.e., long time constant) nature of the SES dynamics. Also, there is a non-minimum phase term in the lateral displacement dynamics such that the ship initially translates in a direction opposite to its final direction for a given rudder command as illustrated in Fig. 3 by the time history of path response to a step change in rudder angle.

---

*"Sub-critical" is used in the manual control context here and must not be confused with the critical speed for the SES. A sub-critical task in the manual control context means that the rate of divergence of the open-loop controlled element as characterized by λ is below the critical level which is at the limit of human manual control capability.

tasks. Participating helmsmen were among SES crewmen with concurrent operational experience from the Surface Effect Ship Test Facility, Naval Air Test Center, Patuxent River, Maryland.

Our purpose here will be to report and interpret some measurements of helmsmen's describing functions during a precision straight-course-keeping task at high cruising speeds in a disturbed sea and involving the use of a compensatory electronic horizontal situation display of heading and lateral displacement errors not specific to any surface effect ship.

APPARATUS AND METHOD

A complement of 3 in. diameter rotary dial instruments provided the helmsman with compass heading, turn rate, rudder angle, and water speed. A visual field simulator provided a collimated external view of the moving seascape and horizon properly correlated with the ship's motions experienced in the cab on the moving base. In addition, an optional automatic helm was provided for relief of manual steering duties while subjective rating analyses were being written and while the helmsman kept watch during four-hour missions.

The random-like steering disturbance was a sum of five non-harmonically related and randomly phased sinusoids whose relative amplitudes were approximately inversely proportional to frequency. This disturbance was generated within the NASA-STI Mark II Describing Function Analyzer (DFA)[4][5] and applied to the mathematical model of the ship's rudder in linear combination with the helmsman's rudder command signal as shown in Fig. 1. The disturbance did not move the helmsman's wheel nor was it directly visible on the helmsman's rudder angle indicator, but its effects on turn rate, heading deviation, and lateral displacement from the desired course were observable. The helmsman was instructed to minimize his lateral error during each 100 sec interval when the DFA was used for cybernetic performance measurement.

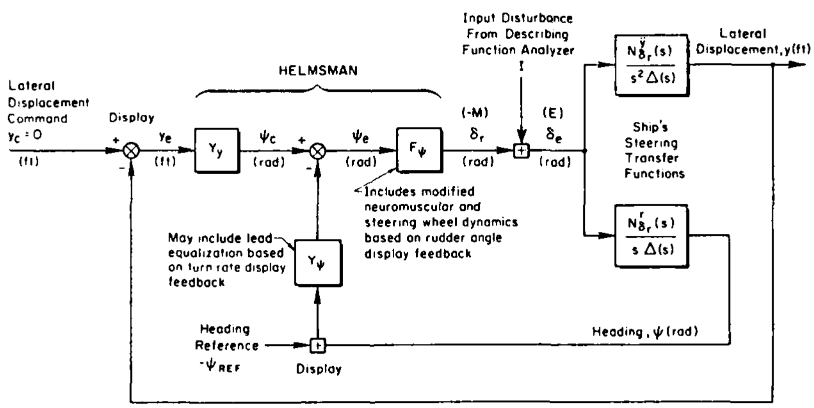


Figure 1. Block Diagram of Heading and Lateral Displacement Control Task as Configured for the Describing Function Analyzer.

The DFA computes on-line the finite Fourier transform, mean-square, and mean of a signal in the control loop — in this case, the rudder error δ_e in Fig. 1 at each of the five input frequencies. The final describing function and error variance are computed off-line with a digital computer program based on some of the techniques in Allen[5]. The principles of the measurement of helmsman/SES

MANUAL STEERING OF A SIMULATED SURFACE-EFFECT SHIP*

by Warren F. Clement
and R. Wade Allen
Systems Technology, Inc.

ABSTRACT

Crew performance in the motion environment of a large generic high speed surface effect ship has been investigated by means of a motion base simulation. Some of the helmsman's control tasks have been addressed with the aid of a simulated external forward visual field of the seascape and navigation and steering displays in the pilot house. In addition to the primary steering control tasks, a subcritical speed tracking task provided a secondary surrogate for trimming the water speed of the ship. The results of helmsmen's steering describing function measurements are presented, and some suggestions for their interpretation are offered. The likely steering loop closures comprise heading and lateral displacement for the course-keeping task investigated. Regardless of the influence of workload, steering technique, water speed, and sea state, the helmsmen apparently adopted a disturbance regulation bandwidth of about 0.2 rad/sec for lateral displacement. Suggestions for reducing the variability in future helmsmen's measurements are offered.

INTRODUCTION

The Surface Effect Ship (SES) is an ocean-going vessel employing a self-generated aerostatic cushion in contact with the water surface for vertical support(1). The SES has rigid shallow-draft side walls with flexible fore and aft skirts or seals to contain the pressurized air cushion while permitting the passage of surface waves through the cushion plenum. The side walls serve as keels to provide lateral stability in the manner of a catamaran. Because the SES rides on a cushion of air, it is less subject to the drag penalties which limit the speed of displacement-hull vessels. Consequently, the SES is capable of sustained higher speeds, and requires precision in course- and sea-keeping, especially in aggravated sea conditions.

The ability of crewmen to perform shipboard duties without undue fatigue or decreased proficiency has been the subject of recent investigations with manned motion base simulation using motion predictions from a 2000-ton SES mathematical model(2)(3). A simulated mission profile with assigned crew tasks provided a disciplined scenario for measuring crew performance in the simulated ship motion environment. Various tasks involving facsimile shipboard operations at four duty stations were performed. One of the duty stations in the pilot house is that of helmsman. In the simulation the helmsman's assignment included a division of attention among steering, speed regulation, obstacle avoidance, and communication

*This research was sponsored by the Naval Sea Systems Command under Contract N00024-73-C-0914.

The authors also gratefully acknowledge significant contributions by their colleagues, Messrs. L. G. Hofmann, H. R. Jex, R. E. Magdaleno, and R. A. Peters of Systems Technology, Inc.

(7) J. Bentson, T. P. Sargent and A. I. Raff, SES Motions and Loads Program "User's Manual," Oceanics Incorporated Report No. 71-85, August 1971.

(8) C. J. Boyd, W. L. Malone, and J. M. Vickery, "Simulation as a Design Aid for Ride Control Systems," Proceedings of the Fourth Ship Control Systems Symposium, October 1975.

(9) J. Schneider and P. Kaplan, "The Incorporation of Fan Dynamics into the Motion Simulation of Surface Effect Ships," Proceedings of the Fourth Ship Control Symposium, October 1975.

(10) P. Kaplan and S. Davis, "A Simplified Presentation of the Vertical Plane Dynamics of the SES Craft," AIAA Paper No. 74-314, February 1974.

SYMBOLS:

CFS = $(\text{feet})^3\ (\text{seconds})^{-1}$

PSF = $(\text{pounds})\ (\text{feet})^{-2}$

ft = feet

tronic complexity. Still, when advanced ship designs are considered or if critical controls are to be applied, a number of potential advantages exists. A few of these advantages have been cited in the preceeding material and and their implementation indicated. A particularly attractive feature of the discribed implementation is the ability to adjust the algorithms by means of weighting functions. Potential approaches for automating the key ship control functions of maneuvering and ride control were also identified. It is hoped that the future will provide the opportunity to try some of these approaches in an ocean going SES.

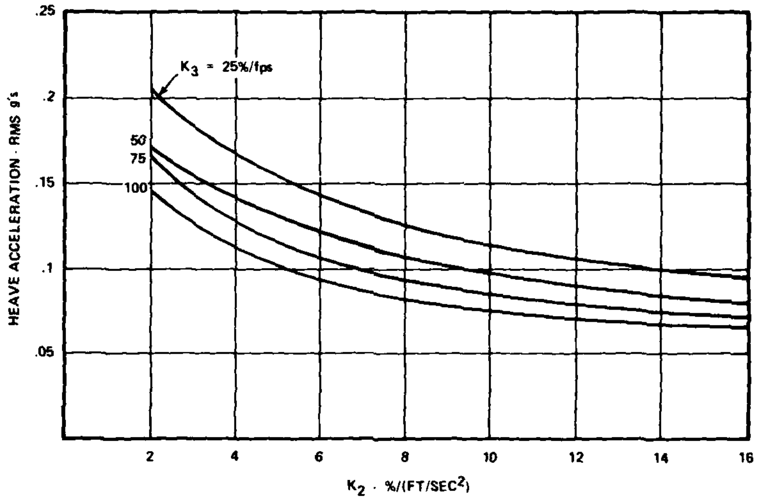


Figure 16. Standard deviation of heave acceleration as a function of controller gains.

REFERENCES

(1) J. Dachos, W. A. Behan, and B. V. Tiblin, "Integrated Bridge System," Proceedings of the Fourth Ship Control Symposium, October 1975.

(2) Rohr Marine Incorporated, "Steering and Reversing Controls Tradeoff Study and Analysis Report," Rohr Marine Incorporated Report DM2S003,501, April 1976.

(3) A. Greensite and L. Pascoe, "3KSES Ship Control System Design Information Report," Rohr Marine Incorporated Report TER023, Vol. I, March 1978.

(4) F. Hauser, "Lift and Ride Control Analysis Report," Rohr Marine, Incorporated Report DL5S00701, August 1976.

(5) F. Hauser, "Preliminary 2KSES Lift and Ride Controller Tradeoff Study Report and Analysis Report," Rohr Marine Incorporated Report DL5S003(5)01, November 1975.

(6) W. C. Foster, D. L. Gieseking, and W. W. Waymyer, "A Non-Linear Filter for Independent Gain and Phase (With Applications)," Journal of Basic Engineering, June 1966.

When a compensation network (Figure 13) with gains K_1, K_2, K_3 and K_4 is included in this same formulation, the resulting equations for these key parameters become:(4)

$$\omega_H^2 = \left(\frac{A_C}{m} + C_3\right)\left(\frac{A_C h_C}{P_T} + C_1\right)^{-1} \tag{17}$$

$$\omega_L = \left(\frac{A_C}{m}\left(\frac{\partial Q}{\partial Z}\right) + C_4\right)\left(\frac{A_C h_C}{m} + C_1\right)^{-1} \tag{18}$$

$$\zeta_H = -\frac{\omega_L}{2\omega_H} + \frac{1}{2\omega_H}\left(\frac{Q}{2\Delta P_C} - \frac{\partial Q}{\partial P} + \frac{h_C}{\gamma P_T}\left(\frac{\partial Q}{\partial Z}\right) + C_2\right)\left(\frac{A_C h_C}{P_T} + C_1\right)^{-1} \tag{19}$$

$$K_0 = \left(\frac{\partial Q}{\partial Z} + K_4\frac{\partial Q}{\partial Z}\right)^{-1} \tag{20}$$

where:

$\frac{\partial Q}{\partial VG}$ = Slope of the VG fans (CFS/%)

$C_j = \frac{A_C K_j}{m}\left(\frac{\partial Q}{\partial VG}\right)$, j= 1, 2,3,4

Based on these relationships the individual gains are seen to have the following effects:

1. As K_1 increases, ω_H decreases
2. As K_2 increases, ζ_H increases
3. As K_3 increases, ω_H increases
4. As K_4 increases, ω_H increases and ω_L decreases.

Such a combination of gains should allow "fine tuning" of the RCS for a given sea state or to assure uniform response over a band of sea states. An example of this tuning is shown in Figure 16 which indicates the standard deviation of heave acceleration as a function of K_3 and K_2 with $K_1 = K_2 = 0$ for a ship with the configuration indicated in Table 1. As is evident, the RCS has the capability to reduce heave acceleration by a factor of two.

Table 1. Parameters for Ship and RCS

PARAMETER	VALUE
A_C	18,790 ft^2
m	166,855 slugs
P_C	254.3 PSF
Q_o	35,000 CFS
$\left(\frac{\partial Q}{\partial P}\right)$	632.6 CFS/%
$\left(\frac{\partial Q}{\partial Z}\right)$	5740 CFS/ft
$h_C A_C$	349,923 ft^3

CONCLUSION

A number of tradeoffs with respect to cost, reliability and maintainability are required before committing to a given level of automation. Based on past experience some of these tradeoffs may not reflect too favorably on the use of increasing elec-

REFERENCES

(1) a) Aronson, R., "Birth of the 100-Knot Navy," Machine Design, Vol. 45, No. 13, May 31, 1973, pp. 20-25.

b) Hudlock, R., "Toward the 100-Knot Navy," Astronautics and Aeronautics, Vol. 11, No. 9, Sept. 1973, pp. 6-10.

(2) Skolnick, A., "Crew Performance Requirements in the Vibration Environments of Surface Effect Ships," Presented at the Session "B" of the AGARD Aerospace Medical Panel Specialists Meeting, Oslo, Norway, 22-25 Apr. 1974.

(3) Jex, H. R., R. J. DiMarco, and W. F. Clement, "Effects of Simulated Surface-Effect Ship Motions on Crew Habitability and Control Operations," Presented at the Fifth Ship Control Systems Symposium, Annapolis, MD, 30 October to 3 November 1978.

(4) Jex, H. R., and R. W. Allen, "Research on a New Human Dynamic Response Test Battery; Part I, Test Development and Validation; Part II, Psychophysiological Correlates," Proceedings of the Sixth Annual Conference on Manual Control, Air Force Institute of Technology, 7-9 Apr. 1970, pp. 743-777.

(5) Allen, R. W., and H. R. Jex, "A Simple Fourier Analysis Technique for Measuring the Dynamic Response of Manual Control Systems," Proceedings of the Sixth Annual Conference on Manual Control, Air Force Institute of Technology, 7-9 Apr. 1970, pp. 785-801.

(6) Teper, G. L., "An Effective Technique for Extracting Pilot Model Parameter Values from Multi-Feedback, Single-Input Tracking Tasks," Proceedings of the Eighth Annual Conference on Manual Control, University of Michigan, 17-19 May 1972, pp. 23-33.

(7) McRuer, D. T., D. H. Weir, H. R. Jex, R. E. Magdaleno, and R. W. Allen, "Measurement of Driver/Vehicle Multiloop Response Properties with a Single Disturbance Input," Proceedings of the Ninth Annual Conference on Manual Control, Massachusetts Institute of Technology, 23-25 May 1973, pp. 217-2[illegible].

(8) Peters, R. A., and R. W. Allen, Operation Manual for Describing Function Analyzer, Model 1003, Serial 1001. Systems Technology, Inc., Working Paper 406-2, Oct. 1970.

SYMBOLS

e	Naperian Base, 2.71828...
E	Error signal output to DFA (I - M)
E/I	DFA error-to-input describing function
F_ψ	Helmsman's heading feedforward describing function including neuromuscular steering actuation dynamics (dimensionless)
I	DFA disturbance input to rudder steering axis
j	$\sqrt{-1}$
K_s	Throttle control-to-speed response gain (ft/sec-rad)
K_ψ	Gain equalization in F_ψ (dimensionless)
M	Closed-loop motion output with respect to DFA input

M/E	Open-loop output to error describing function as measured by DFA
$N^{r}_{\delta_r}$	Controlled element transfer function numerator polynomial representing yaw rate response to rudder displacement (1/sec)
$N^{\ddot{y}}_{\delta_r}$	Controlled element transfer function numerator polynomical representing lateral acceleration response to rudder displacement (ft/sec^2-rad)
s	Laplace operator, $\sigma \pm j\omega$
T_L	Time constant of lead equalization in Y_ψ or F_ψ (sec)
u	Perturbed longitudinal water speed of craft with respect to trimmed speed (ft/sec)
y	Lateral displacement of craft (ft)
y_c	Lateral displacement command (ft)
y_e	Lateral displacement error ($y_c - y$)
Y_y	Helmsman's lateral displacement describing function (rad/ft)
Y_ψ	Helmsman's heading feedback describing function (dimensionless)
δ_e	Rudder angle error (rad)
δ_r	Perturbed rudder deflection angle with respect to trimmed angle (rad)
δ_t	Throttle displacement (rad)
Δ	Characteristic determinant, transfer function denominator
ζ_N	Damping ratio of second-order lag in F_ψ representing effective neuromuscular actuation dynamics modified by proprioceptive or visual feedback (dimensionless)
λ	Inverse time constant of the first-order sub-critical tracking task (rad/sec)
σ	Real part of the complex variable s (rad/sec)
τ_d	Effective helmsman's time delay, including transport, equalization, and scanning contributions (sec)
ψ	Heading of craft (rad)
ψ_c	Heading command ($Y_y y_e$) (rad)
ψ_e	Heading error ($\psi_c - \psi$) (rad)
ω	Circular frequency; imaginary part of the complex variable s (rad/sec)
ω_c	Unit gain crossover frequency (rad/sec)
ω_N	Undamped natural frequency of second-order lag in F_ψ representing effective neuromuscular actuation dynamics modified by proprioceptive or visual feedback (rad/sec)
ω_u	Unstable phase crossover frequency (rad/sec)

ABBREVIATIONS

dB	Decibel
deg	Degree
DFA	Describing Function Analyzer
ft	Foot
in.	Inch
rad	Radian
sec	Second
SES	Surface effect ship
$(\cdot)$	(raised period) Time Derivative Operator d/dt (1/sec)

SYSTEM ANALYSIS TECHNIQUES FOR DESIGNING RIDE CONTROL SYSTEMS FOR SES CRAFT IN WAVES

by Paul Kaplan, Oceanics, Inc.
and Sydney Davis, NAVSEA, PMS-304

ABSTRACT

Procedures for design of a control system (known as Ride Control System or RCS) for reducing the vertical plane accelerations of SES craft by means of altering the pressure variation within the cushion are described. An outline is given of a method using control system analysis techniques in terms of the craft fundamental dynamics in its uncontrolled mode (fundamental frequency and damping characteristics). Particular feedback signals in terms of craft state variables are established in this simplified analysis, which is complicated by saturation limits of the control effector, lags in actuator displacement, etc. (an illustration of a nonlinear saturation limit is the size of the deck leakage area louvers or vent valves used as a typical means of SES control). The analytical model is described and initially applied to louver systems, with and without bias in the area openings. The system analysis technique provides a measure of rms accelerations, demanded control output and a means of obtaining optimal responses as a function of the control gains in a parametric study. Illustrations are given for control via controlled area louver systems; axial fans with variable blade angles; and variable geometry centrifugal fans. The basic system analysis technique illustrated in the present paper have a wide range of applicability to the different possible concepts used for establishing RCS designs for SES craft, and provide the initial gains for establishing an optimal control system.

INTRODUCTION

The surface effect ship (SES) is a seagoing vehicle that operates at high speed at the interface region between air and water, with its major means of weight support obtained from the contained air pressure within a cushion region under the main deck. The concept underlying the SES craft, in contrast to that of other vehicles such as hovercraft, involves the use of rigid sidewalls intersecting the water surface and the presence of fore and aft seals (bow and stern) which act to retain the pressurized air within the so-called cushion region. The resulting motions of this craft when operating in waves are thus primarily due to dynamic effects that arise from the changing pressures associated with the action of the waves in modifying the cushion volume. A simplified description of the basic dynamics that influence the vertical plane motions of SES craft in waves has been presented in[1], with primary emphasis on the resulting heave accelerations of the craft.

The high level of vertical accelerations experienced by SES craft, as indicated from both experimental and computer simulation studies, has illustrated the necessity of incorporating some form of control system for reducing these acceleration levels. The main purpose in reducing these heave acceleration levels is to satisfy the requirements for human operator habitability on board such craft.

While the precise level for SES craft acceleration spectra is being investigated[2] for different missions, the goals for the action of a control system are primarily aimed at reducing the heave acceleration levels to the smallest possible value while still considering the possible penalties associated with the use of such a control system.

In order to obtain an optimal control performance a fundamental investigation has to be carried out in order to consider the effects of different physical means of control, as well as to establish the particular control rules that should be used in order to obtain the best performance. In addition to such considerations, recognization of the realistic limits of actuation of different devices on actual craft must be made together with consideration of lags and other limiting effects. All of these aspects should be investigated in order to determine their influence on the ultimate craft performance. In addition, in any overall investigation of control systems for reducing SES craft heave acceleration, a detailed examination of the power penalty associated with the particular type of control must be made in order to obtain a measure of the overall efficiencies of such control action, as the power requirements are an obvious practical limitation.

The present paper will only consider an initial analysis technique for establishing a control system for SES craft, which serves as a basic approach to define the nature of the control system signal processing associated with different physical means of control, i.e. different control effectors and/or mechanisms. The basic system analysis technique relies upon utilizing knowledge of the fundamental dynamics of the craft vertical plane motion presented in[1] in order to provide the initial gains and control rule for establishing an optimal control system. The information obtained from the method described here can then be used for more detailed studies of system performance with a complete nonlinear six degree of freedom motions program for SES, as described in[3], which can then be used to explore a more precise measure of motion performance for particular craft with different control concepts. In view of the more simple approach based on system analysis that is presented in this paper, no consideration of the power penalty associated with different control concepts is presented since it is beyond the capabilities of that method and is more directly associated with a detailed study using the large computer program.

ASSUMPTIONS AND TECHNIQUES OF ANALYSIS

In carrying out the present analysis a number of simplifying assumptions are made, most of which have been applied in the basic dynamics study of (1) which is used as a starting point in the present work. Since the major concern here is the craft heave acceleration, the craft motion is only represented by the heave degree of freedom which is coupled only with the pressure variable. Linear craft motion equations are used, with the only nonlinearity being due to the control limits or other aspects of the control action. There is no additional leakage due to motions, with the seals assumed to be perfect sealing elements throughout any motion conditions. The only excitation or disturbance action that the control system affects is that due to the wave pumping action, which is evaluated for the head sea case that has the greatest expected heave acceleration response. There are no hydrostatic and/or hydrodynamic forces acting on the craft, with only the

air pressure providing the support force and disturbance reactions. Thus the means of control action must therefore involve some method of altering the air pressure by affecting the air flow characteristics in the cushion.

The method of analysis used is via frequency response methods, together with a quasi-linear method of representing nonlinear effects which will be commensurate with the methods used for analysis of the (primary) linear elements in the system. The SES craft will be assumed to operate at some forward speed in different random sea states represented by the Pierson-Moskowitz wave spectrum, so that stochastic process outputs in statistical form will be obtained. While the major disturbance acting on the craft is due to wave pumping action in the cushion region, some considerations of transient response properties of the controlled system will also be made. The transient response properties will be shown to also influence the response to the wave distrubances, and in addition some discussion will also be given to the responses of the controlled craft due to leakage arising from craft motions (primarily pitch motions).

EQUATIONS OF MOTION

The craft equations of motion are given in [1] in the form

$$m\ddot{z} + A_b p_o \mu = 0 \tag{1}$$

and

$$K_1 \dot{\mu} + K_3 \mu - {}_a A_b \dot{z} = - K_2 \, \Delta A_L - \rho_a \dot{V}_{b_{waves}} \tag{2}$$

where

m	=	craft mass
A_b	=	cushion area
p_o	=	equilibrium cushion pressure (gage)
μ	=	$\frac{p - p_o}{p_o}$, nondimensional pressure variation
z	=	heave motion
ρ_a	=	density of air
ΔA_L	=	change in leakage area (controlled)
K_1, K_2, K_3	=	system parameters from fan characteristics, cushion geometry, equilibrium pressure
$\dot{V}_{b_{waves}}$	=	time rate of change of volume pumping due to waves

This equation system is derived under the conditions of an (effective) linear fan characteristic curve. The fan curve is represented by

$$Q_{in} = Q_o + \left(\frac{\partial Q}{\partial p}\right)_o p_o \mu \tag{3}$$

where Q_o is the fan flow rate when $p = p_o$ and $\left(\frac{\partial Q}{\partial p}\right)_o$ is the assumed linear fan slope (inverse), as indicated in Figure 1.

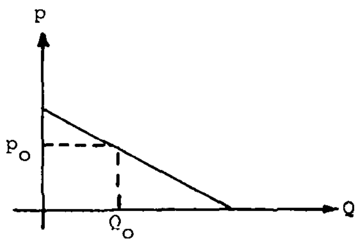


Figure 1. Linear Fan Characteristic Curve.

The parameters K_1, K_2 and K_3 are defined by

$$K_1 = \frac{\rho_a A_b h_b}{\gamma\left(1 + \frac{p_a}{p_o}\right)} \quad , \quad K_2 = \rho_a c_n \sqrt{\frac{2p_o}{\rho_a}}$$

$$K_3 = \frac{\rho_a Q_o}{2} - \rho_a \left(\frac{\partial Q}{\partial p}\right)_o p_o \tag{4}$$

where

h_b = height of cushion

γ = ratio of specific heats for air

p_a = atmospheric pressure

c_n = orifice coefficient

The quantity $\dot{V}_{b_{waves}}$ is defined for a regular sinusoidal head sea wave as

$$\dot{V}_{b_{waves}} = - A_b a \omega_e \frac{\sin \pi \ell/\lambda}{\pi \ell/\lambda} \cos \omega_e t \tag{5}$$

where

a = wave amplitude

ℓ = cushion length

λ = wavelength

and ω_e is the (circular) frequency of encounter given by

$$\omega_e = \omega + \frac{\omega^2 u}{g} \tag{6}$$

with ω = wave frequency and u = craft forward speed.

This particular system of equations is appropriate to the case where additional controlled leakage is provided via louver area variation. However other possible control system implementation concepts can also be represented using these equations, as will be shown later in this paper.

CONTROL SYSTEM ANALYSIS

The design of a useful system for reducing heave accelerations in waves, i.e. a ride control system (RCS), must consider a number of different items of importance. Low values of rms heave acceleration must be achieved together with good dynamic response to impulsive transient disturbances, while not demanding too large a fan power increase. The use of different optional control techniques, either in the frequency or state variable domain, would lead to complicated compensation networks that are obtained from lengthy computations. In addition the particular optional control would vary significantly for different craft operating conditions, i.e. speed-sea state conditions. Also whenever large sea conditions are encountered the entire utility of a linear analysis is subject to questions.

A simple representation of the equations of motion in state variable form is given by

$$\dot{\bar{z}} = A\bar{z} + \bar{b}\, A_L + \bar{c}\dot{V}_{b_{waves}} \tag{7}$$

where

$$\bar{z} = \begin{bmatrix} \dot{z} \\ \mu \end{bmatrix}, \quad \bar{b} = \begin{bmatrix} o \\ b \end{bmatrix}, \quad \bar{c} = \begin{bmatrix} o \\ c \end{bmatrix}$$
$$A = \begin{bmatrix} o & -g \\ a_1 & a_2 \end{bmatrix} \tag{8}$$

and the following definitions apply:

$$a_1 = \frac{\rho_a A_b}{K_1}, \quad a_2 = -\frac{K_3}{K_1}$$
$$b = -\frac{K_2}{K_1}, \quad c = -\frac{\rho_a}{K_1} \tag{9}$$

With this form, and the nature of the coefficient matrices, the representation of a simply-structured control scheme can be achieved in terms of state variable feedback to achieve a desired form of characteristic polynomial equation describing the system. Since the state vector can be easily measured, the control scheme considered is

$$\Delta A_{L_c} = \hat{k}_1 \dot{z} + \hat{k}_2 \mu \tag{10}$$

where

$$\hat{k}_1 = \frac{k_1}{b}, \quad \hat{k}_2 = \frac{k_2}{b} \tag{11}$$

and ΔA_{L_c} is the <u>commanded</u> change in leakage area.

Control Analysis for Louver Control

The technique for control via louvers involves changing the leakage area through variation of an additional opening (in the deck usually). This procedure allows controlled air leakage when the

cushion pressure rises, but there is no way to increase pressure by means of an additional leakage opening. However the most effective means to achieve a useful heave acceleration reduction via louvers is in allowing the area change to occur about some reference condition, so that area variation in both directions, i.e. positive and negative, can be achieved (referred to as 2-sided control). The mean power requirements increase significantly for this type of control, depending on the mean effective position of the louver area opening, i.e. if that value is the same as the base equilibrium leakage area of the craft then the fan power is doubled. Thus it is necessary to limit the extent of the equilibrium louver opening (and hence the total area variation) in order to achieve a reduced additional power due to the control, while still obtaining useful heave acceleration reduction. This is a typical trade-off illustration, whose exact consideration cannot be treated in this simplified analysis but only from some detailed simulation studies (i.e. with the aid of the large digital computer program).

When applying the control rule of Eq. (10) to this case, consideration must be given to the limits imposed by the maximum leakage area permitted in both directions (relative to the equilibrium reference louver area opening). This is illustrated by the symmetrical saturation limit for the actual leakage area, as shown in Figure 2.

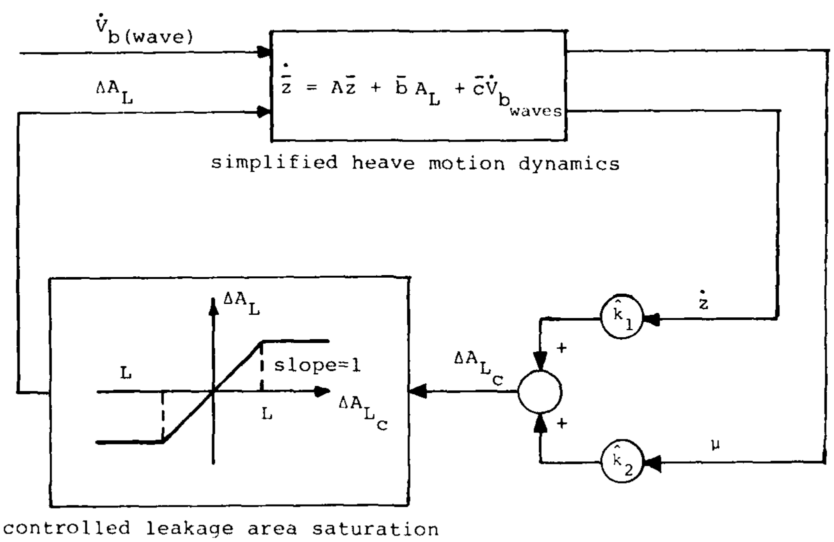


Figure 2. Model of Simplified Heave Motion Dynamics Including Controlled Leakage Area Saturation.

The analysis where the wave inputs arise from a random sea involves a zero-mean, Gaussian stochastic process where the action of the nonlinear limit is represented by the use of a describing function for random inputs. The describing function for this symmetric saturation nonlinearity is a representation as a quasi-linear gain, given by

$$K(\sigma) = \frac{2}{\sqrt{\pi}} \int_0^{L/\sigma\sqrt{2}} e^{-u^2} du = \text{erf}\left(\frac{L}{\sigma\sqrt{2}}\right) \tag{12}$$

where

$$\sigma^2 = \overline{\left(\Delta A_{L_{\text{comm.}}}\right)^2} \tag{13}$$

is the mean square value of the commanded leakage area change.

Using standard Laplace transform notation for the transfer functions, the following relations are obtained:

$$\sigma^2 = \overline{\left(\Delta A_{L_{\text{comm.}}}\right)^2} = \int_0^\infty |T_1(j\omega_e)|^2 \; T_\eta^2(\omega) \; S_\eta(\omega) d\omega \tag{14}$$

$$\sigma_{\ddot{z}}^2 = \overline{\ddot{z}^2} = g^2 \int_0^\infty |T_2(j\omega_e)|^2 \; T_\eta^2(\omega) \; S_\eta(\omega) d\omega \tag{15}$$

where $s = j\omega_e$ is used in the transfer functions $T_1(j\omega_e)$ and $T_2(j\omega_e)$; the quantity $T_\eta(\omega)$ represents the wave pumping volume transfer function in terms of wave frequency ω; $S_\eta(\omega)$ is the surface wave elevation spectrum; and the frequencies are related by Eq. (6). The transfer functions T_1 and T_2 are defined by

$$T_1(s) = \frac{\frac{cs}{b}(k_2 s - k_1 g)}{s^2 - [a_2 + k_2 K(\sigma)]s + [a_1 g + k_1 K(\sigma)]} \tag{16}$$

$$T_2(s) = \frac{cs^2}{s^2 - [a_2 + k_2 K(\sigma)]s + [a_1 g + k_1 K(\sigma)]} \tag{17}$$

and the function $T_\eta(\omega)$ is given by

$$T = \frac{2gA_b}{\omega^2 \ell} \sin\left(\frac{\omega^2 \ell}{2g}\right) \tag{18}$$

It can be seen from the denominators of the transfer functions T_1 and T_2 where the effect of the control gains (k_1 and k_2) in altering the basic natural frequency and damping of the system response is indicated, together with the effect of the nonlinear limit operation in limiting the influence of these parameters. The solution of the above equations requires iteration since the quantity σ entering into the definition of $K(\sigma)$ is not known.

The solutions are obtained for different operating conditions of sea state and ship speed. An effective manner of presenting the results is in terms of dynamic response parameters such as natural frequency and damping, which are obtained in the linear range (when

K( ) = 1) with the definitions

$$\omega_n = \sqrt{(a_1 + k_1)g} \tag{19}$$

$$\zeta = -(a_2 + k_2)/2\omega_n \tag{20}$$

with ζ the damping ratio of the system.

Computations can be carried out for particular cases where the reference louver area opening is some particular fraction of the craft equilibrium base leakage area A_{L_O}. The condition for symmetric action of the louver area variation, as indicated in Figure 2 and the present analysis, corresponds to an opening that is 1/2 of the maximum possible louver area. Thus if the saturation limits are $\pm L$, as shown in Figure 2, the equilibrium reference louver area is L and the area can vary from 0 to 2L. The selection of the value of L is usually referred to the quantity A_{L_O} for ease of analysis.

The computations are carried out for different values of k_1, which would affect the linear natural frequency, with variations of k_2 to produce desired values of linear damping ratio for transient response considerations. All of these computations are carried out for each particular speed-sea state operating conditions in head seas. Results of computations for the louver control, as well as other means of control to be discussed in the following paragraphs, are given in a later section of this paper.

Analysis of Fan Blade Angle Control

Another possible method for reducing heave acceleration is by direct action of the fan, whereby the changes in certain characteristics of the fan would result in changes in the pressure acting in the craft cushion. The particular fan characteristic that can be altered in the case of axial fans is the pitch angle of the blades during fan rotation, which then provides changes in the pressure and flow rate properties of the fan in accordance with the particular blade angle value. Representative fan maps relating pressure as a function of the flow rate for different blade angles are shown in Figure 3, as obtained from curves supplied by a manufacturer of axial fans, which are then used to investigate this control concept.

For purposes of the present analysis, the idealization of these curves is an interpretation whereby the curves have the same basic slope (in the p-Q fan maps) for the different values of blade angle when using an assumed linear fan map. In the basic derivation of Eq. (2), as originally presented in (1), the flow characteristic of the fan inflow to the cushion is represented by

$$Q_{in} = Q_o + \left(\frac{\partial Q}{\partial p}\right)_o p_o \mu \tag{21}$$

where this relationship is assumed to hold for a particular fan blade angle, $\alpha = \alpha_{ref.}$, which is the reference blade angle about which the changes in angle take place. The control of blade angle is assumed to affect only the quantity Q_O in Eq. (21), so that the representation of the fan flow into the cushion is

$$Q_{in} = Q_o \left(1 + \frac{\Delta\alpha}{\alpha_{ref.}}\right) + \left(\frac{\partial Q}{\partial p}\right)_o p_o \mu \tag{22}$$

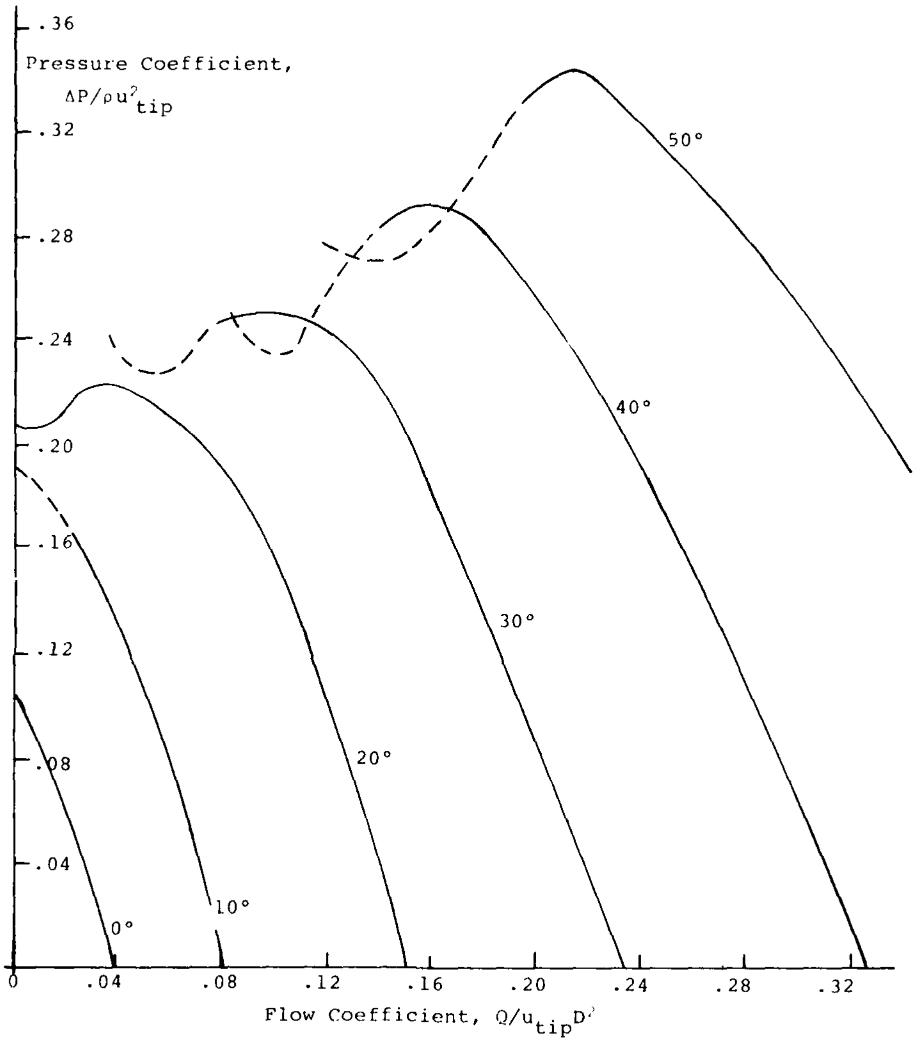


Figure 3. Typical Pressure Variation vs. Flow Rate (coefficient form) for Various Blade Angles, Axial Flow Fans

with $\Delta\alpha$ the change in the fan blade angle. The value of $\Delta\alpha$ is limited such that

$$1 + \frac{\Delta\alpha}{\alpha_{ref.}} \geq 0 \tag{23}$$

and also with an upper limit of $\frac{\Delta\alpha}{\alpha_{ref.}} \leq 1$, leading to

$$-1 \leq \frac{\Delta\alpha}{\alpha_{ref.}} \leq 1 \tag{24}$$

which is the complete limit representation.

Examining the derivation in[1], the representation of the cushion outflow relationship there is

$$\rho_a Q_{out} = \rho_a Q_o + \frac{1}{2} K_2 A_{L_o} \mu + K_2 \Delta A_L \quad , \tag{25}$$

where

$$\rho_a Q_o = K_2 A_{L_o} \tag{26}$$

With the fan flow into the plenum given by Eq. (22) this leads to

$$\rho_a (Q_{in} - Q_{out}) = K_2 A_{L_o} \frac{\Delta\alpha}{\alpha_{ref.}} - K_3 \mu - K_2 \Delta A_L \tag{27}$$

For the control case where there is no commanded leakage (i.e. louver control), the quantity $\Delta A_L = 0$, but it can be seen from Eq. (27) that the quantity A_{L_o} $\Delta\alpha/\alpha_{ref.}$ is essentially equivalent to the controlled leakage area change if the <u>sign</u> of the control gains is reversed. Thus any results obtained by the use of a two-sided louver control, with the ΔA_L limited by $\pm A_{L_o}$, will be exactly the same as from the present fan blade angle except for the sign change of the control gains.

The equations of motion for this case with fan blade angle control result in Eq. (2) being replaced by

$$K_1 \dot{\mu} + K_3 \mu - \rho_a A_b \dot{z} = K_2 A_{L_o} \frac{\Delta\alpha}{\alpha_{ref.}} - \rho_a \dot{V}_{b_{waves}} \tag{28}$$

with Eq. (24) also applied, with the control rule (command signal) given by

$$\left(\frac{\Delta\alpha}{\alpha_{ref.}}\right)_{com.} = -\hat{k}_1 \dot{z} - \hat{k}_2 \mu \tag{29}$$

These equations are then combined in the same manner as for the louver control, using a describing function representation for the symmetric saturation limit action of Eq. (24), and applied to the same case treated there, i.e. the two-sided louver control that is equivalent to

the present problem of fan blade angle control. This control rule is then available as an initial optimal control form to be applied in the system analysis of fan blade angle control, using values of A_{L_o} for a particular SES craft in the specified operating conditions (speed-sea state combinations).

Analysis for Fan Area Control

The use of variable fan areas as a means of control is based on a recent development of such a system for centrifugal fans[4]. The fan inlet area is changed so that the fan output flow is then altered, at almost constant pressure, in order to affect the resultant pressure in the cushion. A general representation of the fan map for fans of this type is shown in Figure 4. The three(3) curves in Figure 4 represent the basic p-Q variation for a particular degree of fan inlet area, denoted as 1/2, 3/4 and fully open, with all intermediate positions also possible. If the condition corresponding to 3/4 open is considered as a reference base area, the possible changes in area to 1/2 or full open correspond to the relation

$$-\frac{1}{3} \le \frac{\Delta A}{A_{ref}} \le \frac{1}{3} \qquad (30)$$

where ΔA is the change in fan inlet area.

Since the curves in Figure 4 indicate that the slope of the fan maps (p-Q curves) also changes with the area change, a fit of these curves was found to be represented (approximately) by

$$Q = (Q_o - \Delta Q\cdot\mu)\left(1+1.5\frac{\Delta A}{A_{ref.}}\right) \qquad (31)$$

where the first parenthesis term represents the flow rate characteristic for the fan case are of 3/4 open (the reference condition in the present case). Other possible numerical values would be present for different fan curves, but the present case is sufficient for illustrative purposes.

Since the action of the fan is to change the input flow to the cushion, and the leakage (louver) control is a means to change the outflow, the control action can be represented similarly with only a change in sign for the command gains. The change in the basic pressure equation leads to

$$K_1\dot{\mu}+K_3\mu-\rho_a A_b\dot{z}=1.5K_2A_{L_o}\frac{\Delta A}{A_{ref.}}+1.5\rho_a\left(\frac{\partial\phi}{\partial p}\right)_o\left(\frac{\Delta A}{A_{ref.}}\right)-\rho_a\dot{V}_{b_{wave}} \qquad (32)$$

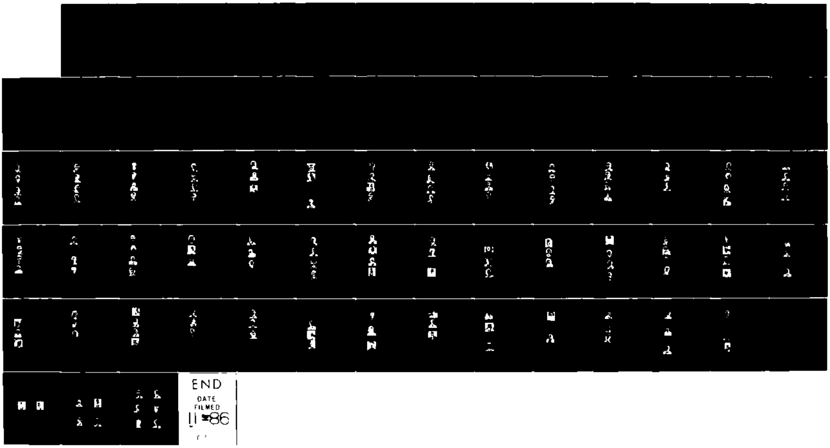
where A_{L_o} is equilibrium leakage area, with the control command signal given by

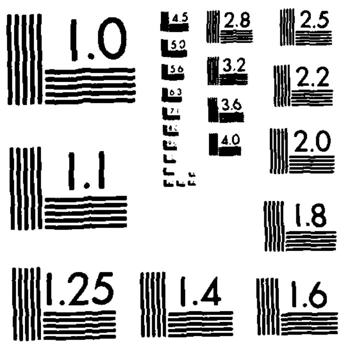
$$\frac{\Delta A}{A_{ref.}} = -\hat{k}_1\dot{z} - \hat{k}_2\mu \qquad (33)$$

subject to the limit given by Eq. (30).

AD-A159 086 PROCEEDINGS OF THE SHIP CONTROL SYSTEMS SYMPOSIUM (5TH) HELD AT U S NAVAL..(U) DAVID W TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER ANN.. E G BURT ET AL. 03 NOV 78 2/2

UNCLASSIFIED F/G 13/10 NL





'CROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

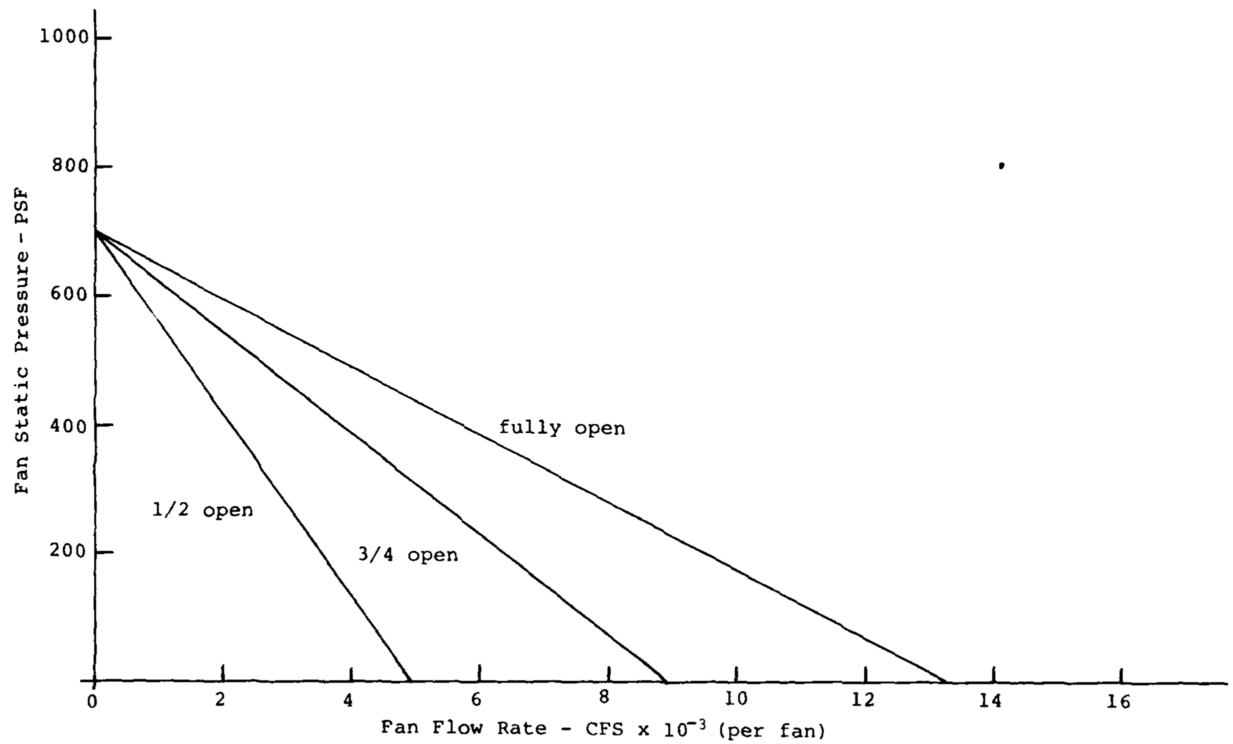


Figure 4. Variable Area Fan Characteristics

The problem here is to find the values of the gains $\hat{k}_1$ and $\hat{k}_2$ to provide an optional control for this case. It can be shown that the nonlinear product $\dot{z}\mu$ that results from Eq. (32) has a negligible contribution (since $\mu \sim \dot{z}$ the resulting product has no contribution to a describing function value) and can be neglected. Similarly the μ^2 product resulting from the nonlinearity in Eq. (32) can be shown to have an influence of the order of 10% on the equivalent linear representation of the control input, when neglecting the effect of the saturation limit action. This treatment is based upon the assumption of an equivalent linearization of the μ^2 term by use of the mean value of μ that occurs for SES craft (a small negative value, indicating deeper draft in waves), as well as avoiding consideration of the complexities arising from multiple nonlinearieites in a describing function analysis. On the basis of the above reasoning the nonlinearity due to the product expression in Eq. (32) is ignored (since a 10% change in the gain factor is not significant within the level of accur ·y of the present analysis), while the nonlinear limit action indicated by Eq. (30) is still retained. The effect of such a treatment can only be accurately assessed by means of a more complete simulation study, which also includes other effects neglected in this overall analysis, so that the results obtained here are only to be viewed as initial estimates of control gains for extended simulation studies.

Based on the analysis discussed above, the actual term entering the equation system is

$$1.5A_{L_o} \frac{\Delta A}{A_{ref}} = -1.5A_{L_o} (\hat{k}_1 \dot{z} + \hat{k}_2 \mu) \tag{34}$$

and this term is bounded by $\pm \frac{1}{2}A_{L_o}$ when considering the action of the control as a symmetric variation about the 3/4 area reference.

When considering the operation of the variable area fans as two separate groups, i.e. a biased control with half the total fans operating at 1/2 open and the other half of the fan system operating at full open, the equations are changed somewhat. In that case the 1/2 open fan can operate (under control) only to increase the inlet area and operate up to the limit of the full open condition, i.e. to allow an increase in flow into the cushion, while the full open fans can operate under the control to reduce the flow into the cushion, down to the 1/2 open condition. This operation involves knowing an equation for the fan curve for each operating condition, and then the resulting equation for the change of flow rate under the actions of the control applied to each operating fan group. In that case the flow in can be represented by

$$\begin{aligned} Q_{in} &= \left[Q_{o_{1/2}} - \left(\frac{\partial Q}{\partial p}\right)_{o_{1/2}} p_o \mu\right] \left[1 + C_1\left(\frac{\Delta A}{A_{ref}}\right)_{1/2}\right] \\ &+ \left[Q_{o_1} - \left(\frac{\partial Q}{\partial p}\right)_{o_1} p_o \mu\right] \left[1 + C_2\left(\frac{\Delta A}{A_{ref}}\right)_1\right] \end{aligned} \tag{35}$$

where the 1/2 and 1 subscripts co-respond to the operating fan group (area opening value), with the control action given the limit constraints

$$\left(\frac{\Delta A}{A_{ref}}\right)_{1/2} > 0 \ , \ \leq 1$$

$$\left(\frac{\Delta A}{A_{ref}}\right)_{1} < 0 \ , \ \left|\frac{\Delta A}{A_{ref}}\right| \leq \frac{1}{2} \tag{36}$$

These limit actions are represented by asymmetric saturation nonlinearities, with the 1/2 area reference case being a positive saturation limit and the 1 reference condition being a negative saturation limit.

The resulting equations contain the values of $Q_{O1/2}$, Q_{O1}, etc. which can be combined to provide effective values of $\hat{k}_3$, a representation of the effective equilibrium leakage areas $A_{L_{O1/2}}$ and $A_{L_{O1}}$, etc.

These equations can be simplified somewhat by elimination of the nonlinear product terms $\mu\dot{z}$ and μ^2, as discussed previously, and also simplified by means of a control action that requires

$$A_{L_{O_{1/2}}} C_1\left(\frac{\Delta A}{A_{ref}}\right)_{1/2} = A_{L_{O_1}} C_2\left(\frac{\Delta A}{A_{ref}}\right)_1 = -C_3\,(\hat{k}_1\dot{z} + \hat{k}_2\mu) \tag{37}$$

The values of C_1 and C_2 are determined from the actual fan curves in the "fit" prescribed by Eq. (35) in such a way that the effective control action is then made symmetrical, i.e. a symmetric saturation limit as shown in Figure 2. For the case considered previously, in terms of the fan maps given in Figure 4, it can be shown that the effective representation of the control input can be given by

$$A_{L_O}\left(\frac{\Delta A}{A_{ref}}\right) = -A_{L_O}(\hat{k}_1\dot{z} + \hat{k}_2\mu) \tag{38}$$

with the limits

$$-1 \leq \frac{\Delta A}{A_{ref}} \leq 1 \tag{39}$$

This particular control input is then included in the equations in the same manner as the case for variable blade angle variation for axial fans, with the term $\Delta\alpha/\alpha_{ref}$ in Eq. (28) replaced by $\Delta A/A_{ref}$ for the present case of variable area centrifugal fans. Calculated results obtained in both of these cases, assuming the same ship, A_{L_O} value, operating conditions, etc. will yield the same numerical value for controlled responses when using the same k_1 and k_2 values.

RESULTS OF COMPUTATIONS

Computations were carried out in order to determine the resulting heave accelerations for different operating conditions (i.e. speed-sea state combinations) for a representative 2000 ton SES design. The computations covered a louver system with the total deck area opening extending up to the equilibrium leakage area value A_{L_O} (approximately 87.5 ft^2), as well as the two cases of fan control via changing blade angle (axial fans) or inlet area (centrifugal fans).

The results for the louver control can be used to provide values for the responses and associated gains for the case of variable area

fans operating symmetrically (about the 3/4 open condition, as expressed in Eq. (30)-(34)), since the range of the limit given by Eq. (34) is $\pm 1/2\, A_{L_o}$. The signs of the gains have to be reversed for the fan area control case, as indicated previously. For the biased fan operating conditions, i.e. half the fans at 1/2 open and the other at full open condition for the reference conditions, the results are obtained subject to the limit given by Eq. (28) and (29), which is the same as the case for axial fan blade angle control represented by Eq. (28) and (24).

The computations were carried out for an operating range of a representative 2000 ton craft in head seas, at a speed of 40 kt. in Sea State 4; a speed of 50 kt. in Sea States 4 and 5; and a speed of 60 kt. in Sea States 4 and 5. The results obtained are given in Tables 1-5 for the louver control case, as well as for the symmetric fan area control case, for the range of louver area opening up to $\pm 1/2\, A_{L_o}$ and the fan area change $\Delta A/A_{ref}$ in the range $\pm 1/2$. The values of the gains k_1 and k_2 given in those tables apply to the louver control, where

$$A_{L_{comm}} = \frac{1}{b}\,(k_1\dot{z} + k_2\mu) = 52\,(k_1\dot{z} + k_2\mu) \qquad (40)$$

For the symmetric fan area control the results are represented by

$$A_{L_o}\,\frac{\Delta A}{A_{ref}} = -\,52\,(k_1\dot{z} + k_2\mu) \qquad (41)$$

using the same values of k_1 and k_2 tabulated in Tables 1-5, with essentially the same expected heave acceleration values. Similar considerations apply to the case of the "biased" fan operation, with the larger limit condition on fan area change, as well as the axial fan blade area control, with the results in those cases given in Tables 6-10.

Table 1. Results of Control Analysis, Louver and Symmetric Fan Area Control

u = 40 kt., Sea State 4
(uncontrolled runs heave accel. = 0.20g)

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	13.604	0.0878	- 7.796
	1	17.261	0.0533	-10.328
	2	20.934	0.0394	-12.208
	3	23.225	0.0317	-13.773
1.0	0	18.106	0.0670	-12.882
	1	19.450	0.0422	-17.947
	2	22.031	0.0320	-21.706
	3	23.883	0.0261	-24.836
1.5	0	21.062	0.0545	-17.968
	1	21.532	0.0351	-25.565
	2	23.310	0.0270	-31.204
	3	24.748	0.0223	-35.899

Table 2. Results of Control Analysis, Louver and Symmetric Fan Area Control

u = 50 kt., Sea State 4
(uncontrolled rms heave accel. = 0.25g)

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	14.673	0.1084	- 7.796
	1	17.654	0.0681	-10.328
	2	21.106	0.0512	-12.208
	3	23.327	0.0416	-13.773
1.0	0	19.115	0.0814	-12.882
	1	20.126	0.0527	-17.947
	2	22.416	0.0405	-21.706
	3	24.132	0.0334	-24.836
1.5	0	21.965	0.0656	-17.968
	1	22.335	0.0432	-25.565
	2	23.855	0.0337	-31.204
	3	25.137	0.0280	-35.899

Table 3. Results of Control Analysis, Louver and Symmetric Fan Area Control

u = 50 kt., Sea State 5
(uncontrolled rms heave accel. = 0.31g)

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	29.825	0.1725	- 7.796
	1	47.006	0.1211	-10.328
	2	75.166	0.1148	-12.208
	3	105.292	0.1145	-13.773
1.0	0	45.635	0.1560	-12.882
	1	59.244	0.1133	-17.947
	2	85.744	0.1068	-21.706
	3	114.975	0.1061	-24.836
1.5	0	61.147	0.1484	-17.968
	1	75.220	0.1116	-25.565
	2	101.088	0.1044	-31.204
	3	129.854	0.1027	-35.899

Table 4. Results of Control Analysis, Louver and Symmetric Fan Area Control

u = 60 kt., Sea State 4
(uncontrolled rms heave accel. = 0.27g)

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	15.559	0.1292	- 7.796
	1	18.033	0.0834	-10.328
	2	21.277	0.0636	-12.208
	3	23.426	0.0521	-13.773
1.0	0	19.918	0.0958	-12.882
	1	20.736	0.0635	-17.947
	2	22.782	0.0493	-21.706
	3	24.374	0.0410	-24.836
1.5	0	22.663	0.0767	-17.968
	1	23.021	0.0513	-25.565
	2	24.290	0.0404	-31.204
	3	25.430	0.0339	-35.899

Table 5. Results of Control Analysis, Louver and Symmetric Fan Area Control

u = 60 kt., Sea State 5
(uncontrolled rms heave accel. = 0.38g)

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	32.568	0.2118	- 7.796
	1	48.673	0.1524	-10.328
	2	76.533	0.1453	-12.208
	3	106.598	0.1455	-13.773
1.0	0	49.887	0.1924	-12.882
	1	63.329	0.1431	-17.947
	2	89.499	0.1352	-21.706
	3	118.576	0.1344	-24.836
1.5	0	67.223	0.1844	-17.968
	1	82.074	0.1416	-25.565
	2	107.945	0.1327	-31.204
	3	136.662	0.1305	-35.899

Table 6. Results of Control Analysis, Biased Fan Area Control (2 groups) and Axial Fan Blade Angle Control

u = 40 kt., Sea State 4

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	13.598	0.0878	- 7.796
	1	17.150	0.0530	-10.328
	2	20.389	0.0383	-12.208
	3	22.153	0.0302	-13.773
1.0	0	17.958	0.0664	-12.882
	1	19.144	0.0415	-17.947
	2	21.263	0.0308	-21.706
	3	22.617	0.0247	-24.836
1.5	0	20.553	0.0530	-17.968
	1	20.883	0.0340	-25.565
	2	22.216	0.0257	-31.204
	3	23.197	0.0208	-35.899

Table 7. Results of Control Analysis, Biased Fan Area Control (2 groups) and Axial Fan Blade Angle Control

u = 50 kt., Sea State 4

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	14.657	0.1083	- 7.796
	1	17.518	0.0675	-10.328
	2	20.529	0.0497	-12.208
	3	22.226	0.0396	-13.773
1.0	0	18.874	0.0802	-12.882
	1	19.727	0.0516	-17.947
	2	21.556	0.0388	-21.706
	3	22.787	0.0315	-24.836
1.5	0	21.270	0.0633	-17.968
	1	21.506	0.0415	-25.565
	2	22.599	0.0318	-31.204
	3	23.446	0.0261	-35.899

Table 8. Results of Control Analysis, Biased Fan Area Control (2 groups) and Axial Fan Blade Angle Control

u = 50 kt., Sea State 5

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	28.487	0.1623	- 7.796
	1	37.435	0.0943	-10.328
	2	46.276	0.0689	-12.208
	3	51.991	0.0553	-13.773
1.0	0	38.565	0.1275	-12.882
	1	42.153	0.0778	-17.947
	2	48.684	0.0582	-21.706
	3	53.478	0.0473	-23.836
1.5	0	45.531	0.1056	-17.968
	1	46.939	0.0666	-25.565
	2	51.663	0.0507	-31.204
	3	55.541	0.0417	-35.899

Table 9. Results of Control Analysis, Biased Fan Area Control (2 groups) and Axial Fan Blade Angle Control

u = 60 kt., Sea State 4

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	15.527	0.1289	- 7.796
	1	17.869	0.0826	-10.328
	2	20.666	0.0617	-12.208
	3	22.296	0.0495	-13.773
1.0	0	19.577	0.0940	-12.882
	1	20.238	0.0618	-17.947
	2	21.830	0.0471	-21.706
	3	22.950	0.0385	-24.836
1.5	0	21.797	0.0735	-17.968
	1	22.018	0.0491	-25.565
	2	22.935	0.0380	-31.204
	3	23.673	0.0314	-35.899

Table 10. Results of Control Analysis, Biased Fan Area Control (2 groups) and Axial Fan Blade Angle Control

u = 60 kt., Sea State 5

ζ, damping ratio	k_1	σ	rms heave accel., g's	k_2
0.5	0	30.525	0.1950	- 7.796
	1	38.136	0.1165	-10.328
	2	46.595	0.0862	-12.208
	3	52.191	0.0697	-13.773
1.0	0	40.606	0.1509	-12.882
	1	43.455	0.0945	-17.947
	2	49.434	0.0716	-21.706
	3	53.977	0.0586	-24.836
1.5	0	47.477	0.1241	-17.968
	1	48.589	0.0799	-25.565
	2	52.789	0.0616	-31.204
	3	56.362	0.0510	-35.899

Comparing the results obtained by these different methods, it can be seen that the greatest heave acceleration reduction occurs when using the variable area fan in the biased manner, i.e. as two separate groups, or by use of the axial fans with variable blade angle control. This is due to the larger range of effective "area" change possible in that case, since the saturation effect at half that range (for the louver or symmetric variable area fan) limits the benefits of the other techniques. Of course similar benefits could be obtained by using a larger range of louver area, with an attendant larger louver area bias level, but that would immediately demand greater lift system power. The best control system must really be evaluated via more detailed computer simulation, where power requirements are also determined, but the present results do indicate the expected region of benefit so that the required detailed computer simulation studies of such control effects can be limited to the more promising types of control concept.

Examination of the results given in the tables shows that in general the rms heave accelerations are reduced further as the gains k_1 and k_2 are increased. However, in some results obtained in other studies the effect of the saturation is exhibited, so that an optimum value for that particular operating case is evident. The tables also indicate the degree of difference due to the particular value of the damping ratio, with the greater acceleration reduction indicated for the largest value of damping ratio. However good dynamic response to transient disturbances is required and too large a value of ζ would give a sluggish transient response. Since the linearized damping ratio would not be fully attained throughout the motion time history of the controlled craft, the most suitable value of damping ratio should be $\zeta = 1.0$ for satisfactory transient response.

While the values tabulated in the tables indicate a significantly large reduction of the rms acceleration, it is not expected that such effects would be realized in actual practice (or even with the large scale digital computer simulation). This is due to the presence of other forces acting on the craft that have been neglected, but more important is the effect of leakage arising from the craft motions. Such leakage is expected to occur primarily due to pitch motion that

leads to leakage gaps at the bow due to the relative motion with respect to the sea surface at that location. The present analysis does not consider these important effects, but it is expected that the aim of designing for adequate transient response, and also with an effective increase in the craft natural frequency, would be beneficial in reducing the responses due to impulsive disturbances that arise due to transient leakage. The main use of the data presented in the tables is therefore to establish a control rule, with initial gain value estimates, for use in the necessary large scale digital computer simulation studies for arriving at an optimum RCS for SES craft in waves.

FULL SCALE TESTCRAFT RESULTS WITH RIDE CONTROL SYSTEM

While the computational results given in the tables apply to a representative 2000 ton SES design, such craft are not presently built and operating. However 100 ton testcraft have been operating for some time in order to obtain data that can be used as a means of predicting the expected characteristics of full scale behavior, as well as functioning as a test-bed for evaluating various concepts that could be applied to multi-thousand ton SES craft in the near future.

Some illustrative data that demonstrate the effectiveness of the RCS system using the louver-type control mechanism is shown in Figure 5. In that case the time histories show a specific illustration of the general reduction in heave acceleration and cushion pressure when the RCS is activated. Further detailed information that illustrates the benefit obtained from the use of an RCS system is given by Figures 6 and 7, where power spectra of heave acceleration and cushion pressure variations, both with and without the RCS system operating, are shown. These illustrations of actual full scale performance of 100 ton testcraft with RCS are indicative of the nature of the benefits that can be obtained with such a system. Continued analysis and prediction using the initial methods described in this paper, together with more detailed large digital computer simulation studies, will allow development of an optimal RCS for use in the presently developing multi-thousand ton SES craft.

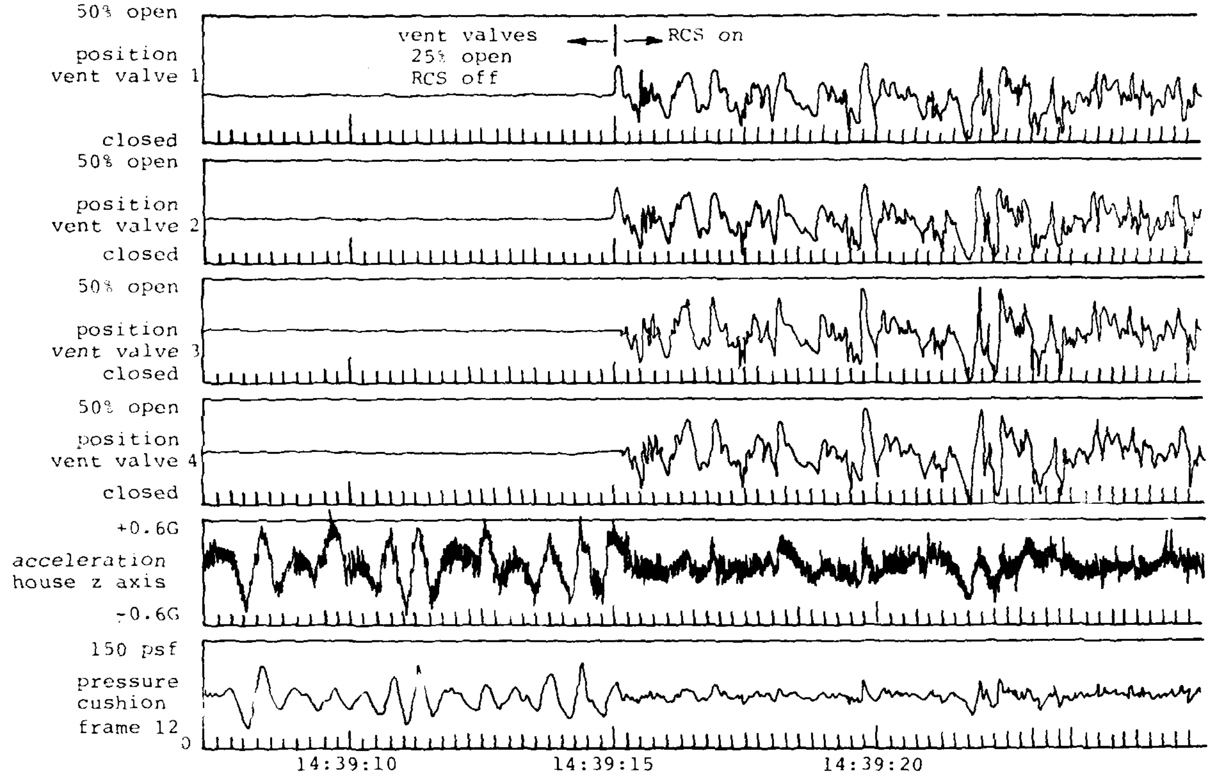
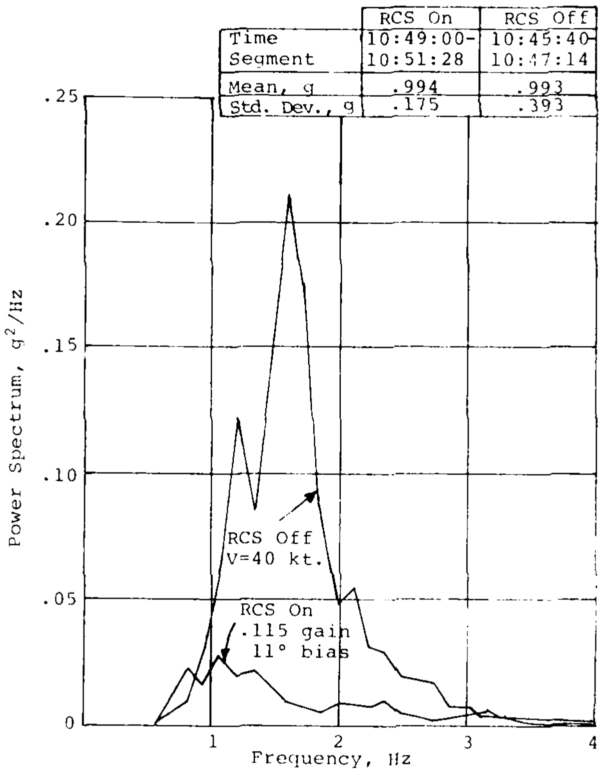


Figure 5. Time Histories of RCS Action for 100-Ton Testcraft in Head Seas, Avg. Ht. 3 ft., Speed 35 kt.



	RCS On	RCS Off
Time Segment	10:49:00-10:51:28	10:45:40-10:47:14
Mean, g	.994	.993
Std. Dev., g	.175	.393

Figure 6. Mission 111 Power Spectrum of $\ddot{z}$ (cabin), Head Sea State 2, RCS On and Off.

	Volume	Session	Page
Shipley, P. Birkbeck College (UK)	5	Q1	3-1
Simanowith, R. C. David W. Taylor Naval Ship R&D Center	5	R	2-1
Smalley, J. P. E. EASAMS, Ltd (UK)	4	O1	1-1
Smith, W. E. David W. Taylor Naval Ship R&D Center	6	P	2-1
Spencer, J. B. Chairman, Session K2 Ministry of Defence (UK)	1	A	2-1
Stankey, R. Chairman, Session E1 Naval Ship Engineering Center			
Steinhausen, J., LCDR, RN Ministry of Defence (UK)	4	O1	1-1
Stuurman, A., LT Royal Navy (Neth)	2	E1	3-1
Sugimoto, A. Mitsubishi Heavy Industries, Ltd (Japan)	4	P	1-1
Thomas, J. R. E. Ferranti (UK)	2	E1	2-1
Thompson, R. V. Univ of Newcastle Upon Tyne (UK)	5	Q2	2-1
Tiano, A. Laboratorie per l'Automazione Navale (Italy)	4	P	4-1
Toney, J., LT, USN Naval Postgraduate School	3	K2	1-1
Turner, R. J. College of Nautical Studies (UK)	4	O1	3-1
Turner, T. Vosper Thorneycroft, Ltd (UK)	2	F1	2-1

	Volume	Session	Page
Hindmarsh, G. J. Vosper Thornycroft, Ltd (UK)	5	R	3-1
Hoffman, D. Hoffman Maritime Consultants, Inc.	4	P	3-1
Holland, G. E. Naval Ship Engineering Center	1	A	1-1
Horigome, M. Tokyo Univ of Mercantile Marine (Japan)	1	C	2-1
Houlihan, T. M. Naval Postgraduate School	3	K2	1-1
Joby, M. J. Lucas Aerospace, Ltd (UK)	2	G	3-1
Johnson, R. David W. Taylor Naval Ship R&D Center	3	J1	3-1
Kallstrom, C. Lund Inst of Tech (Sweden)	3	J2	2-1
Kammerer, J. G. Naval Ocean Systems Center	3	J1	1-1
Kaplan, P. Oceanics, Inc.	2	F2	4-1
Karasuno, K. Kobe Univ of Mercantile Marine (Japan)	2	D	3-1
Keating, G. Decca Radar, Ltd (UK)	2	F1	3-1
Kern, D. Chairman, Session L Specialized Systems, Inc.			
Keyes, P. R. MARA-TIME Marine Services Corp.	3	K1	1-1
King, Randolph W., RADM (Ret) Speaker, Symposium Dinner Maritime Transportation Res Bd Natl Res Council			

LIST OF SYMPOSIUM AUTHORS, SESSION CHAIRMEN,
AND GUEST SPEAKERS

REFERENCES

(1) P, Kaplan, and S. Davis, "A Simplified Representation of the Vertical Plane Dynamics of SES Craft," AIAA Pape No. 74-314, AIAA/SNAME Advanced Marine Vehicles Conference, San Diego, CA., February 1974.

(2) C.J. Boyd, W.L. Malone, and J.M. Vickery, "Simulation as a Design Aid for Ride Control Systems," Proc. Fourth Ship Control Systems Symposium, Royal Netherlands Naval College, October 1975.

(3) J. Bentson, P. Kaplan, and S. Davis, "Simulation of Surface Effect Ship Motions and Loads," Summer Computer Simulation Conference, Washington, D.C., July 1976.

(4) Aerojet Liquid Rocket Company, "Lift Fan System for the U.S. Navy 2000 Ton Surface Effect Ship Program," Booklet No. MSD 2200LF742, 1974.

	RCS On	RCS Off
Time Segment	10:49:00-10:51:28	10:45:40-10:47:14
Mean, psf	88.5	92.1
Std. Dev., psf	15.2	38.9

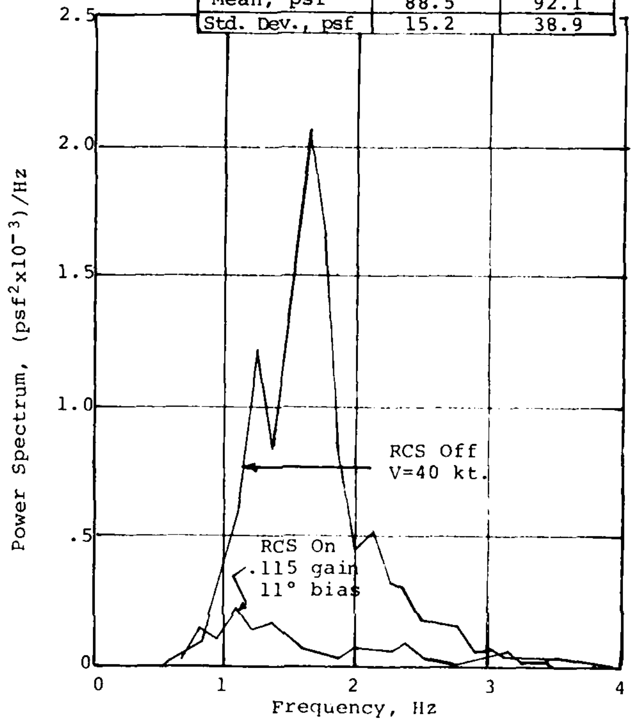


Figure 7. Mission 111 Power Spectrum of Cushion Pressure (Frame 12), Head Sea State 2, RCS On and Off.

SYMPOSIUM AUTHOR BIOGRAPHIES

TALLAK AAS born in Norway August 11, 1943. He was educated at the Technical University of Trondheim. His experience has been in Project and R&D work in the field of maritime computerized control/training simulators.

R. WADE ALLEN received the B.S. and M.S. degrees in engineering from the University of California at Los Angeles in 1962 and 1965, respectively. He has been actively involved in human factors and man/machine systems research since 1960. As a member of the technical staff at Hughes Aircraft Company from 1964 to 1967 he was involved in research on a wide range of manual control system problems, in addition to more discrete tasks requiring detection and recognition by the human operator. Since 1967 he has been with Systems Technology, Inc., Hawthorne, California, where he is presently a senior research engineer. Mr. Allen is a member of the Human Factors Society and IEEE.

G.E. BAAS completed his study in electronics at the Technical College in Rotterdam. After fulfilling his military service he joined TNO-IWECO in 1969. During several years he was involved in analog simulation of ships and in manual control studies, using the IWECO-Ship Bridge Simulator. In 1974 he became Head of the digital-computing facility.

TREVOR C. BARTRAM was born in Hull, England on 4 April 1946. He is a graduate of the University of Bradford, gaining the degree of B.Tech. Hons. in Electrical and Electronic Engineering. Mr. Bartram joined International Research and Development Company Ltd. in 1969. His interest has extended in the design of d.c. superconducting machines and systems in addition to other engineering applications of superconductivity. Mr. Bartram was appointed Head of the Electrical Engineering Department at IRD in 1977.

A. BAXTER obtained a BSc (Mathematics) at Strathclyde University (Glasgow) in 1972 and MSc (Mechanical Engineering) 1973 also from Strathclyde. Since graduating he has worked in the Machinery Control Section at National Gas Turbine Establishment (West Drayton), formerly the Admiralty Engineering Laboratory, on computers and their application to the control of ships' machinery. He has recently completed the degree of M Tech (Computer Science) at Brunel University (Uxbridge).

Adrian Birbănescu-Biran graduated in 1952 with a Diploma of Ingenieur, in Naval Architecture and Marine Engineering, from the Bucharest Polytechnic Institute. In 1977 he obtained the title of Master of Science from the Technion-Israel Institute of Technology, with a thesis on the propulsion dynamics of a gas turbine ship with C. P. propellers. Presently, he is employed at the Israel Shipyards, in Haifa, as a senior R&D engineer. At the same time, he works at the Technion as an adjunct lecturer, teaching Ship Hydrostatics and supervising projects in Ship Design.

MOGENS BLANKE was born in Copenhagen in October 1947. He received the M.S.E.E. degree (Automatic Control) in 1974 from the Technical University of Denmark (D.t.H.), Lynghy. After graduation and until late 1977 he studied toward the Ph.D at Servolaboratory (D.t.H.) and the Danish Ship Research Laboratory specializing in design of minimum loss autopilots for ships. Since 1977 he has been an assistant professor at Servolaboratory working at application of computer control by using microprocessor technology.

THOMAS L. BOWEN is a native of the Chesapeake tidewater area of Maryland. Mr. Bowen earned a Bachelor of Science degree in Chemical Engineering in 1973 from the West Virginia Institute of Technology. Subsequently, he was employed as a full-time Chemical Engineer in the Gas Turbines Branch - DTNSRDC. He is a member of the American Institute of Chemical Engineers and has published several technical papers related to marine gas turbines and alternative fuels.

PAMELA J. BOZZI received her BA in Mathematics Education from the University of Maryland, College Park in 1973, graduating Magna Cum Laude. She worked for ManTech of New Jersey Corporation from 1974 to 1977. From 1977 to the present she has worked for ORI, Inc. Here she was involved in the development of the steering control system for the DDS 963 Class destroyer which is discussed in this paper.

A.W. BRINK graduated in aeronautics at the Delft University of Technology. He was employed by the Royal Dutch Airlines from 1963 until 1970. At that time, he joined TNO-IWECO, which is the Dutch abbreviation for the Institute for Mechanical Constructions of the Org. for Applied Scientific Research. Since then, he has been involved in simulation of ships, and particularly in studies on automatic control of ships during course or track-keeping, but also during station-keeping.

DR. D.R. BROOME is Lecturer in Control Engineering, Mechanical Engineering Department at University College London. Current research interests include an integrated ship control system to account for the multivariable nature of ship dynamics, submarine control systems and adaptive ships autopilots.

C.J. BRUCE joined the staff of the National Gas Turbine Establishment (West Drayton), formerly the Admiralty Engineering Laboratory, in 1969 and worked in the Diesel Engine Test Section. In 1970 he left the Establishment to obtain a BSc in Mechanical Engineering from the Polytechnic of Central London. After graduating in 1973 he returned to the Machinery Control Section of NGTE(WD) and has worked on projects concerned with ship motion control and the application of processor based systems to the control of ships' machinery. He is currently completing an MSc in Instrumentation and Systems Technology from the Polytechnic of Central London.

LENNART BYSTRÖM was born in Bollnäs, Sweden in 1945. He received his MS degree (Civilingenjor) in technical physics in 1970 from the Royal Institute of Technology (KTH) in Stockholm, Sweden. After working with traffic control systems at the Royal Swedish Air Force he is currently employed as a research engineer at the Swedish State Shipbuilding Experimental Tank, Gothenburg. His main research interests are in the field of ship maneuvering.

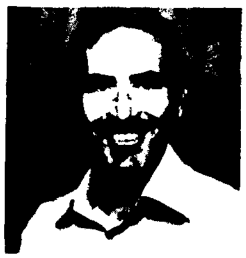
CDR JAMES F. CARRUTHERS BEng. Phd, CD Canadian Forces joined the Royal Canadian Navy in 1961 and obtained a Phd in Electrical Engineering from Nova Scotia Technical College in 1971. In 1974 Carruthers served as project officer for a mini command and control system entitled ADLIPS. It was also during this period that the SHINPADS concept was first put forward. His present responsibilities include all aspects of C^2 systems, shipboard computers, ADLIPS project management, and development of SHINPADS and its associated data bus, standard display, and AN/UYK 502 microcomputer.

CHARLES W. CASSEL obtained his AB degree in Mathematics from Wittenberg University in 1943. Since 1962 his assignments with the General Electric Space Division have included a ship automation cost-benefits study, analysis of FFG-7 Class automatic propulsion controls and math modeling for two FFG-7 propulsion system trainers.

Y.T. CHAN received his Ph.D. from the University of New Brunswick, Fredericton, in Electrical Engineering. He has been with Northern Electric Co. LTD., London, Ontario and Royal Military College of Canada, Kingston, Ontario and Bell-Northern Research, Ottawa, Ontario. Since 1973 he has been a research associate at the Royal Military College, Kingston conducting research on weapons systems for the Directorate of Maritime Combat Systems, Department of National Defence, Canada.

WARREN F. CLEMENT was awarded the B.S. and M.S. degrees in aeronautical engineering from Massachusetts Institute of Technology in 1950 and 1951, respectively. Since 1966 he has been a Principal Research Engineer with Systems Technology, Inc., Hawthorne, California, engaged in man-machine systems research for aircraft and surface effect ship applications. Mr. Clement holds six patents, is a registered control systems engineer, and is a member of both the IEEE Systems, Man, and Cybernetics Society and the American Institute of Aeronautics and Astronautics.

EDWARD M. CONNELLY received a BSEE from Carnegie Institute of Technology in 1953. With the Quest Research Corporation his research was directed to the development of a computer processor to develop pilot performance measures from flight test data. With Omnemii, Inc. he performed research in non-linear operator modeling technology. At Performance Measurement Associates he has worked to develop state transition models of fire growth and of representing human behavior by the criteria apparently optimized. Mr. Connelly is the author of numerous technical papers and reports pertaining to the measurement and modeling of human operators.



J.E. COOLING B.Sc. (Hons), C. Eng., M.I.E.E. specialized in naval electronic control system design for the Control and Simulation Division of Marconi Radar Systems Limited (Leicester) and is now a Research Fellow at Loughborough University of Technology. He is a member of the Institution of Electrical Engineers.

RICHARD B. COOPER is a human factors analyst with Eclectech Associates, Inc. He has extensive experience in human factors analysis, primarily in the marine field. Recently, he has been involved in a human factors analysis for the standardization of merchant ship automation and bridge design, has participated in the National Maritime Research Center CAORF simulator validation, and has investigated design alternatives for a ship's service speed measurement system. He is currently collecting data on the performance of ship's pilots as a function of various new navigation display concepts and on changes to the U.S. Coast Guard aids to navigation.

R.R. CUMMINGS MA RCNC joined the Royal Naval Engineering Service (now the Royal Corps of Naval Constructors) in 1972, having read Engineering and Electrical Sciences at Churchill College, Cambridge. During his employment in the Ministry of Defence he has been concerned with a wide cross-section of prime mover and propulsion control systems and is presently the Project Officer for the Brecon Class main machinery control and surveillance system.

SYDNEY DAVIS received the degree of B.Sc (Eng.) (Hons.) from London University. After teaching engineering for several years he joined the staff of Aerojet General Corporation, as a technical specialist in Hydrodynamics. In this capacity he became a member of the SES team set up by Aerojet. In 1967 he joined the Joint Navy/Marad Surface Effect Ships Program as head of the Aero-Hydrodynamics Branch of the Technology Division. He has been responsible for developing technology in the areas of SES performance, stability and maneuvering, loads, motions and fans. He has written several papers on SES related subjects and has patents pending for improvement of lightweight variable geometry high performance fans.

HENRY J. DEMATTIA is presently a program manager in the Electromagnetics and Acoustics Division (SEA 034) of the Naval Sea Systems Command. He received a B.S. Degree in Mathematics from West Virginia Wesleyan College and Graduate Studies at the George Washington University. Prior to current position with NAVSEA, he was a Ship Design Project Coordinator (electronics) for the Naval Ship Engineering Center (NAVSEC), and also held engineering positions with RCA Service Company, Alexandria, VA and International Electronics Engineering Incorporated, Washington, DC.

JAMES W. DONNELLY Senior Project Engineer was born in Philadelphia, PA on June 20, 1944. He received B.S. and M.S. degrees in electrical engineering from Drexel University in 1967 and 1969, respectively. From 1969 to present, he has been employed in the Machinery Automation Systems Department of the Naval Ship Engineering Center, Philadelphia Div. He was made co-recipient of the NAVSECPHILA-DIV Technical Achievement Award in recognition of the technical support that he provided in the development and evaluation of the control system for the FFG-7 prototype propulsion system.

A.M. DORRIAN is a Chartered Engineer and Member of the Institution of Mechanical Engineers. In 1968 he obtained an H.N.C. with distinction in Mechanical Engineering and in 1970 graduated from Strathclyde University with an Honours Degree in Mechanical Engineering. He completed engineering apprenticeship with G&J Weir Ltd, joined Y-ARD Ltd in 1970 and, following work on a variety of acoustic and vibration problems joined the Controls Group where he has been responsible for the development of the machinery controls philosophy for a variety of naval and special purpose ships. He is currently Head of the Surface Ship Control Systems Section.

KÅRE HARALD DRAGER holds the M.Sc. Electrical Engineering (1966) and M.Sc. Industrial Engineering (1973). Following studies of Ship Automation with computers reliability analysis of bridge instrumentation and systems, he is currently Project Manager of a team studying "Cause Relationship of Collisions and Groundings" for Det norske Veritas.

ARTHUR E. DRESSER obtained the BSME in 1965 and is a Registered Professional Engineer in the State of Maine. He is presently a Senior Mechanical Engineer with Bath Iron Works with responsibilities including Shipboard Automation Project Engineering.

ALBERT DUBERLEY served an apprenticeship in Marine Engineering. He is a chartered engineer and a member of the Institution of Mechanical Engineers. Joined Admiralty Engineering Lab (now National Gas Turbine Establishment) in 1954 where he is now employed as a Principal Scientific Officer and Head of the Machinery Control Section.

HARUZO EDA is a Senior Research Engineer at Davidson Laboratory and a Research Associate Professor for the Department of Ocean Engineering, Stevens Institute of Technology. Employed at Davidson Laboratory since 1961, Dr. Eda is actively engaged in research on stability and control of ships and underwater bodies. Dr. Eda serves as a Member of the Panel H-10 (Ship Controllability) of the Society of Naval Architects and Marine Engineers; a Member of the Maneuvering Committee of International Towing Tank Conference and of the American Towing Tank Conference, and also as a Member of SNAME and the Society of Sigma Xi. Dr. Eda holds Bachelor and Doctoral Degrees from Osaka University in Japan, and a Master's Degree from Stevens Institute of Technology.

EIVIND KINDEN ENGEBRESTSEN, a 1956 graduate from the University of London, has been associated since 1965 with the control and instrumentation section of the Ship Research Institute of Norway. Leader of the section since 1973, Mr. Engebrestsen is in charge of design specification and commissioning of ship automation systems.

R.J. EVES graduated in Electronic and Electrical Engineering at University of Birmingham, U.K., 1965. Has been employed by Lucas Aerospace since 1969 and has been engaged in systems design and analysis of hydromechanical and electronic control systems for Gas Turbine Engines for Aircraft, Industrial and Marine applications. Present position - Performance Project Engineer.

JAMES A. FEIN graduated in 1969 from the University of Michigan with a B.S.E. in Engineering Mechanics. He has been employed at the Naval Ship Research and Development Center from 1969 to the present. At this time he is a naval architect in the Ship Performance Department concerned with the dynamics of high performance vehicles. He received an M.S. degree in Mechanical Engineering in 1972 from the University of Maryland where he is currently pursuing a PhD in the same field. He is a member of American Institute of Aeronautics and Astronautics and the Society of Naval Architects and Marine Engineers.

RAY FERANCHAK, after graduation from Youngstown University, started work at Westinghouse, developing systems and controls for d-c drives for towboats, dredges, and icebreakers. Since then, he has been involved with electrical drives for wind tunnels, missile pumps, aircraft generators, and transit vehicles. He has done graduate work in industrial control systems, servo-mechanisms, and feedback control systems. He is a member of IEEE and has published several papers on aircraft generator test drives.

ALAN S. FIELDS obtained B.E.E. from the City College of New York and a M.E.E. from Drexel Institute of Technology. He has been employed at the David W. Taylor Naval Ship Research and Development Center since 1964, involved in the design of steering, tracking and station-keeping control systems for surface ships.

CDR EUGENE E. FITZPATRICK, USN is a 1959 graduate of the U.S. Naval Academy and holds the MSME and degree of Naval Engineer from MIT. CDR Fitzpatrick served in the Naval Reactors program from 1966 to 1973, served four years in ship overhaul management in the Naval Sea Systems Command and is presently Project Coordinator, Machinery Division, Naval Ship Engineering Center.

FLOYD FOLTZ has been with Hamilton Standard for the past 16 years. He is presently a Project Manager responsible for the FT-9 Engine Condition Monitor and for Hamilton Test Systems TRENDS systems. His major programs have been data acquisition systems and control systems, which provided a background of sensor applications, data acquisition techniques, computer interfaces, and field evaluation. Mr. Foltz was with the U.S. Air Force for seven years.

C. FRENCH served an electrical apprenticeship at a Royal Dockyard meanwhile obtaining a B.Sc. in Electrical Engineering. Following a period of Industrial and University research was awarded a Ph.D. in Engineering. Undertook design work on large turbo-alternators with a power station plant manufacturer, then re-entered the academic world on the staff of a Polytechnic followed by a University. Rejoined the MOD in the Ship Department six years ago, spending the last two as Head of the Systems Development Group in the Machinery Controls Section.

IRA FRIEDMAN received his B.S. degree in Physics at Brooklyn College. Over the past 25 years he has held positions of responsibility in navigation, guidance, and control of air and surface platforms, and in Combat System Integration for the 3K SES (Surface Effect Ship). He was a member of the NSIA Committee investigating new ASW Weapon Concepts for high speed ships. He joined Ingalls Shipbuilding Division, Litton Systems, Inc. in 1977 where he has been involved in Propulsion Control, Ship Data Multiplexing, and Advanced Combatant studies. He holds a U.S. Patent and is a member of the IEEE.

HANS FRIVOLD graduated from the University of Oslo in 1973 with a degree in Applied Mathematics Hydrodynamics. From 1973 to 1977 Mr. Frivold's work was in propeller hydrodynamics, the pressure field due to working propeller. This past year his work has been with mathematical models for ship maneuvering simulation.

DANIEL H. GEORGE received the B.S.E.E. degree from Louisiana State University in 1968. He has worked for the Tennessee Valley Authority at the Browns Ferry Nuclear Power Plant in Athens, Alabama. Since 1970 he has been employed by the Panama Canal Company Engineering Division as a design engineer, and is presently a general engineer with wide responsibility for major studies and projects. Mr. George is a member of IEEE, Tau Beta Pi, and a Registered Professional Engineer in the Canal Zone.

F. WAYNE GIBSON attended the Royal Military College of Canada in Kingston, Ontario, where he obtained a B.Eng. in Engineering Physics in 1971. He has been a Naval Officer in the Canadian Armed Forces since being commissioned in 1971 and has served as a Combat Systems Engineering Officer in HMCS KOOTENAY and a Project Systems Engineer at National Defence Headquarters, Ottawa, Ontario.

CECIL N. GOFF holds a BME from the University of Florida and the MSE from Pennsylvania State University. Since 1968 Mr. Goff has been employed at the Naval Training Equipment Center where he has worked with a Periscope View Simulator, a DD963 Class Propulsion Trainer and a 1200 PSI Propulsion Plant Trainer.

FABIO R. GOLDSCHMIED holds the degrees of BS and MS in Mechanical Engineering from Columbia University. For the past ten years Mr. Goldschmied has been an Advisory Scientist in the Research and Development Center, Energy Systems Division, Westinghouse Electric Corp. He is the author of numerous technical papers for the AIAA Journals of Aircraft and Hydronautics,an Associate Fellow of AIAA and a member of Technical Committee on Marine Systems and Technologies.

SIGURD GÖRANSSON graduated in Electrical Engineering in 1959 from Chalmers University of Technology. In 1964 he joined the staff of the Swedish State Shipbuilding Experimental Tank, where he is in charge of the Hybrid Computer Facilities.

LINDSAY GORRELL obtained the B.A.Sc. degree in Mechanical Engineering from the University of Waterloo in 1969 and the M.Sc. degree in Ergonomics from the University of London in 1970. Mr. Gorrell is presently employed as a Defence Scientist by the Defence and Civil Institute of Environmental Medicine in Toronto. Much of his work involves human engineering evaluation and design of electronic display systems. He is a member of the Human Factors Society and the Human Factors Association of Canada.

LYMAN GRISWOLD graduated from Duke University (BSEE 55) and received a Master of Science in Engineering from George Washington University in 1962. He recently retired from Federal Service as Head of the Sensors Branch at the David W. Taylor Naval Ship Research and Development Center. He is a Captain in the United States Navy Reserve and a contributor to Bowditch (The American Practical Navigator) on the subject of speed logs.

EIGIL HAALAND graduated in 1967 from the Norwegian Institute of Technology. Following six years of research work at the Institute Mr. Haaland joined Norcontrol where he worked on marine control systems. Since 1978, he has been an R&D section leader, instrumentation department.

DONALD D. HALL is presently assigned to the Tactical Computer System Architecture Division at the Naval Ocean Systems Center since 1975, where he has served as a microcomputer network system design engineer. He received his BSEE degree from the University of the Pacific and MSEE from the University of California.

MAKOTO HARA was born in Yamaguchi, Japan, in 1951. He graduated at Dept. of Electronics, Hosei Junior College in 1971. From 1971 to 1972, he was engaged in Kitashiba Electric Co. Ltd. He has been with Dept. of Control Engineering, Tokyo University of Mercantile Marine since 1972, where he is presently a research assistant. His main interest is the application of microprocessors to practical marine control systems.

THURMAN R. HARPER is a Senior Simulation Analyst with Propulsion Dynamics, Inc., Annapolis, Maryland. He has participated in and directed propulsion dynamics and control simulation studies of gas turbine, steam, diesel and combined ship propulsion plants. He has also been employed at the David W. Taylor Naval Ship Research and Development Center, Annapolis engaged in ship automation, control and mathematical model development of gas turbines and ship systems. Dr. Harper holds the B.S.E.E. degree from Drexel University, Philadelphia and the Ph.D. degree in Electrical Engineering from the Johns Hopkins University, Baltimore.

GRAHAM JOHN HINDMARSH B.Sc., M.B.C.S. obtained a first class honours degree in computer science from the Council for National Academic Awards while studying at Wolverhampton Polytechnic U.K. For the last 3 years Mr. Hindmarsh has been Manager of the Computing and Simulation Department of the Products Division of Vosper Thornycroft (UK) Limited. In this capacity he is responsible for the use of computers in engineering applications with particular emphasis on marine machinery control, and for the computer simulation of ship hull and machinery performance leading to the definition of an optimum control specification.

DR. DAN HOFFMAN received his undergraduate degree in Naval Architecture from the University of Strathclyde in Glasgow in 1959. He then proceeded with post graduate work in the Technion, Israel Institute of Technology and the Technical University of Norway. He has been a Research Professor at Webb Institute of Naval Architecture for over ten years and President and Technical Director of Hoffman Maritime Consultants Inc. (HMC) over the past five years. Dr. Hoffman has authored numerous papers in the area of vessel response to the environment and system approach to operations in rough weather.

GEORGE E. HOLLAND received the Bachelor of Science degree from the U.S. Merchant Marine Academy and Bachelor of Mechanical Engineering degree from the University of Virginia. Following active duty in the Navy as Main Propulsion Assistant on USS FREMONT (APA-44) he has been employed as a mechanical engineer by Esso Research & Engineering Co., Chief Engineer by Renick & Mahoney, Inc., and Resident Engineer by Parker Aircraft Co. before joining Machinery Systems Division of the Naval Ship Engineering Center in 1966. Since July 1975 he has been Head, Project Coordination Office.

MICHIO HORIGOME was born in Miyagi, Japan in 1940. He received the B.S. degree in Marine Engineering from Tokyo University of Mercantile Marine in 1963. Since 1965 he has been with the Dept. of Control Engineering, Tokyo University of Mercantile Marine, where he is presently an Associate Professor. He received the Annual Award in 1977 from the Marine Engineering Society in Japan. His research interests are in time series analysis, stochastic process and the application of these methods to Marine Engine Systems.

DR. THOMAS M. HOULIHAN is a native of Buffalo, NY. He received his BSME from Manhattan College and his Ph.D. from Syracuse University. Since 1969 he has been teaching fluid mechanics and instrumentation in the Mechanical Engineering Department of the Naval Postgraduate School. He has received the Teetor Award of the SAE and the Dow Award of the ASEE for his teaching endeavors.

M.J. JOBY graduated in Mechanical Engineering at University College, London 1962. Has since specialized in Control Systems ranging from powered artificial limbs to supersonic engines. Has worked for Lucas Aerospace since 1966. Present position - Performance Manager (Electronics).

IR. H. SCHUFFEL received his degree of Naval Architect at Delft University. Following military service he has been since 1970, research engineer at the Institute for Perception TNO and in charge of the ship-manoeuvring simulator.

RAYMOND C. SIMANOWITH received the degree of BS Mathematics, Drexel University and MS Numerical Science, Johns Hopkins University. He was employed at the David W. Taylor Naval Ship Research and Development Center (DTNSRDC) from 1964-1978 performing as a Computer Systems Analyst. His areas of expertise include system design, software design and data base design.

JONATHAN SMALLEY graduated BSc from Durham University and MSc from Birmingham University. He joined EASAMS Ltd. as a Senior Systems Analyst in 1976 with main interests in the design of Man/Machine Interfaces. He is currently working on the Man/Machine Interface design study, which is the subject of this paper, with responsibility for human factors and related design aspects.

BRIAN SPENCER graduated in 1950 with an honours degree in mathematics. He became a member of the Scientific Advisers group in the Ship Department at Bath in 1965 with particular responsibility for automatic control systems and hydrodynamics. Has been very much involved with this series of Ship Control Systems Symposia, presented a paper at the First, organized the Third, a committee member for the Fourth, co-author of a paper to be presented at the Fifth, and U.K. coordinator.

C. JOSEPH RUBIS is President of Propulsion Dynamics, Inc., Annapolis, Maryland where he is engaged in simulation and analytical studies of ship propulsion and automation systems. Previous to his present position he was a Scientific Consultant and Head of the Central Systems Branch, David W. Taylor Naval Ship Research and Development Center, Annapolis Laboratory, for over 10 years. Mr. Rubis received his MS degree from the University of Illinois. He was the technical co-chairman of the Second Ship Control Systems Symposium in 1969.

LCDR JAMES K. RULAND, USN graduated from Iona College and received his Master's Degree in Business Administration from Pace University. He has served on the USS TACONIC (AGC-17), USS PERRY (DD 844), USS LEAHY (CG-16) and USS NASHVILLE (LPD-13). He also served as officer in charge of PCF-65 in Vietnam in 1968. He is currently assigned to the staff of the Deputy Chief of Naval Operations (Manpower). Recently he was awarded the Meritorious Service Medal for his work as CNO Project Officer for the HARDMAN effort.

ERNST SCHUBERT is presently a scientific assistent in the Display-department of the Forschungs-institut fur Anthropotechnik (FAT). He is currently responsible for the development of integrated electronic prediction displays for ship control. In the last year he studied the adaptation process of the human eye depending on intensity and color of the light in real situation experiments about which he reports in this paper. Ernst Schubert studied experimental physics and received his Diplom-Ingenieur at the TUB (Technical University of Berlin). He is a member of the DPG (German Physical Society), DGaO (German Society of applied Optics) and LiTG (German Lighting Technics Society).

PATRICIA SHIPLEY was educated at Southampton and London Universities and at Birbeck College. She has since spent 10 years as a lecturer in the Department of Occupational Psychology at Birbeck College. She is co-editor of a book on work design and work satisfaction to be published early in 1979, an ordinary member of the Ergonomics Society and currently an elected member of council of that body. She is also an associate member of the British Psychological Society and a founder member of the Division of Occupational Psychology of the BPS.

ROBERT J. RISBERG received the BSCE degree from Notre Dame University in 1948. He went to work for Panama Canal Company immediately upon graduation from College. He has advanced through progressively more responsible management and engineering positions and at present is Chief, Engineering Division for the Panama Canal Company. Mr. Risberg is a Registered Professional Engineer in the Canal Zone and in the State of Illinois.

DR. ECKHARD ROHKAMM received the degree of Dr. Ing. from Technische Universitat. His experience includes four years of active duty in the FGN and six years as a Senior Research Engineer at Technische Universitat. He is presently head of the Scientific Department of the Naval Engineering Branch, Blohm + Voss AG, Hamburg

CAPTAIN (E) OYSTEIN RONNING is a graduate of the Royal Norwegian Naval Academy and received the degree of BSc from the Royal Naval College, Greenwich, U.K. His experience includes service in patrol boats and frigates and in various positions within the Bureau of Ships. He is presently Leader of the Propulsion Systems Division of Bureau of Ships new construction department.

OLAV ROPSTAD was educated at the University of Trondheim, Norway, the Institute for Engineering Cybernetics, receiving the Master of Science degree in 1975. Since 1976 he has been employed by A/S KONGSBERG VAPENFABRIKK, Kongsberg, Norway as a projectengineer at the Department for vessel automation systems. There he has been engaged in development and testing of the system software of Kongsberg Dynamic Positioning System.

THOMAS H. PUTMAN was born in Pittsburgh, Pennsylvania. He received the B.S. degree from Union College, Schenectady, New York in 1952, and the S.M. and ScD. degree from Massachusetts Institute of Technology, Cambridge, Massachusetts in 1954 and 1958, respectively. Upon completion of his work at M.I.T., Dr. Putman joined the Westinghouse Research Laboratories as a Senior Engineer in the Electromechanics Department and subsequently became manager of that activity in 1967. In 1974 Dr. Putman was appointed Consulting Engineer in the Electronic Div. of the Westinghouse Research Laboratories. Dr. Putman's technical activities encompass the analysis and design of electrical, mechanical and hydraulic systems.

DR. DEAN A. RAINS performed his undergraduate and graduate study at the California Institute of Technology where he culminated his education by receiving his PhD in Mechanical Engineering and Mathematics in 1954. During the period 1954 through 1970, he has held numerous positions of responsibility in private industry. In 1970, he joined Litton Ship Systems where he was involved in the design of the LHA and DD-963 marine power plants, and is currently Director, Advanced Engineering Programs for Ingalls Shipbuilding Division, Litton Systems, Inc.

CAPTAIN P. REEVES graduated from the Guided Weapon Post Graduate course at the Royal Military College of Science at Shrivenham in 1962 and worked on weapon servo-control systems 1962-65 with the Director General Weapons. Since 1968 has had a variety of appointments in Director General Weapons and Director General Ships, mainly concerned with control systems. After serving as Weapons Electrical Officer in HMS GLAMORGAN from 1973-75, took up his current post as Head of the Machinery Controls Section in Director General Ships. Promoted Captain December 1977.

ROBERT E. REID received the BSME from the University of Glasgow and the MSME and PhD degrees from the University of Virginia.He is presently a Research Section Head with Sperry Marine Systems, Charlottesville, VA responsible for system design of navigation, steering and control system for military and commercial applications. Mr. Reid is a Chartered Engineer U.K. and a member of the Institution of Mechanical Engineers.

IAN WATSON PIRIE was educated in Scotland and completed a Student Apprenticeship with the Ministry of Defence. After post graduate training he joined the Admiralty Surface Weapons Establishment and worked on the First of Class commissioning and sea trials of Sea Dart missile system. He moved to Bath in 1975 and since then has been involved in a major research and development program into machinery controls and surveillance. He is now the Project Leader for MOD(PE) in the development of a new digital Propulsion Control System.

J.B. PLANT attended Royal Military College of Canada, Kingston, Ontario and R.N.E.C., Plymouth, England, graduating as a Lieutenant (E) in Marine Engineering in 1957. He received his Ph.D. from M.I.T. in Electrical Engineering (Optimal Control) in 1965 and has held an academic appointment in Electrical Engineering at RMC since that time: as Head of the Electrical Engineering Department from 1967 to 1971 and as Dean of Graduate Studies and Research since 1971.

NEIL PROBERT is a graduate of Durham University with a BSc (Hons) in Engineering. He joined Hawker Siddeley Dynamics Engineering in 1975 and was involved in the development of the Propulsion Control System for a new class of corvettes for the Royal Danish Navy from initial design stage through to production testing. Currently, he is working on a prototype micro-processor based propulsion control and surveillance system.

JOSEPH J. PUGLISI in his current capacity manages projects in the areas of computer simulation, electronic navigation and communications at the NMRC Computer Aided Operations Research Facility. He is presently manager of the CAORF facility at Kings Point. His prior experience was as an engineer at the Naval Applied Science Laboratory in Brooklyn and the Naval Ship Research and Development Laboratory at Annapolis.

JOHN N. ORTON was born in 1922. He served for almost thirty years in the Royal Navy. He left the Navy in 1967 with the rank of Lieutenant Commander and joined EASAMS in 1968. Currently he is responsible for a number of Naval studies including the study of the Man Machine Interface requirements for the Command and Control of Ships' Machinery on which this paper is based.

LCDR LARRY E. PARKIN graduated in 1968 with a Bachelor of Science degree and an Ensign's Commission from the U.S. Coast Guard Academy. Previous to his present position at Coast Guard Headquarters, LCDR Parkin's assignments include two tours afloat as a Naval Engineer, post graduate schooling resulting in a Master of Science - Electronics Engineering (MSEE) degree and a tour at the Electronics Engineering Laboratory in Alexandria, Virginia. LCDR Parkin's present position includes responsibility for the design, acquisition, installation, integration and support of the electronics suites for new cutters.

ALAN J. PESCH is President of Eclectech Assoc. Inc. He received a BS in Psychology and an MS in Cybernetics from the University of Wisconsin. He has been directly responsible for the development and implementation of many new concepts employing simulators in marine systems training. He is currently performing research on the man-machine interfaces in the shipping industry for the U.S. Maritime Administration.

LCDR M.A. PHELPS was born and educated in Cheltenham, Gloucestershire. He obtained a B.S. (Mechanical Engineering) at the Royal Naval Engineering College. LCDR Phelps has served as watch keeping section officer on HMS FEARLESS, Deputy MEO of HMS PENELOPE and as Base Engineer Officer. He is currently serving on HMS SULTAN.

NILS H. NORRBIN was born in Nykoping, Sweden in 1926. He holds the Civilingenjor (Naval Architecture), Tekn. License and Tekn. Dr. From 1950 to 1954 he was with the Submarine Design and Preliminary Ship Design Departments of the Royal Swedish Naval Administration. Since 1955 he has been with the Research Department of the Swedish State Shipbuilding Experimental Tank as Head of the Ship Dynamics Section.

KOHEI OHTSU was born in Osaka, Japan on July 3, 1943. He graduated in the navigation course of the Tokyo University of Mercantile Marine (TUMM) in 1967. From 1975, he has been an assistant professor in the TUMM and from 1977, he has held simultaneously the captain of the TS.SHIOJI MARU. His research interests are in the ship's steering control using the time series theory and the marine traffic engineering. He is a member of SNAJ, Japan Institute of Navigation and Japan Statistical Society.

M. OKUMURA is a captain of training ship owned by Kobe University of Mercantile Marine and holds a captain's License unlimited. He is an assistant professor of the University and responsible for the training of students.

B.M. OLSON obtained his B.S. in Marine Engineering and Naval Architecture and a M.S. in Marine Engineering from the Massachusetts Institute of Technology. He is employed at Gibbs & Cox, Inc. where he has served in various capacities including Senior Propulsion Engineer, Head of the Machinery Propulsion Department and Project Engineer. His prior experience with gas turbine propulsion systems includes the H.S. VICTORIA, the PG 84 Class, the RCN DDH 280 Class and as consultant to Litton Industries on the DD963 Class. He has been active in the dynamic analysis of marine propulsion systems for over ten years.

RICHARD F. MESSALLE graduated from St. Vincent College with a BA in Mathematics and received a MS in Operations Research from George Washington University. He has been with the High Performance Craft Dynamics Branch at DTNSRDC since 1974. He is a member of the Mathematical Association of America.

JOHN MOON received B.A. (Open University), Dip. Tech. (Brighton College of Technology), M.Sc. and Ph.D. (Birmingham University) degrees. He worked in the British nuclear power industry from 1963 to 1966 and then returned to academic life doing research into neutron transport theory. Since 1973 he has worked for Ferranti Computer Systems Limited and is currently a Group Leader of the System Problem Analysis Group. He is a member of the Institute of Physics and an Associate Fellow of the Institute of Mathematics and its Applications.

DAVID D. MORAN was awarded a Doctorate in Hydrodynamics by the University of Iowa (Ph.D., 1971). He has been employed by David W. Taylor Naval Ship Research and Development Center since 1971 where he has been engaged in ship powering and resistance research for the Ship Performance Department Powering Systems Branch; and, Advanced Vehicle Dynamics Research for the High Performance Craft Dynamics Branch. Dr. Moran is presently acting manager of the Ship Performance Department Exploratory Development Programs Office.

DONALD G. MOSS is a graduate of Kansas State College. He has worked for the General Electric Company for 22 years. For the last eight years he has worked on the development of machinery controls for Navy Ships, currently assigned to the AO-177 Fleet Oiler controls program. He is at present senior systems engineer with the Ground Support Department at Daytona Beach, Florida.

PATRICK MARTIN received his Engineering degree from the "Institut Superieur d'Electronique de Paris" in Paris in 1971. In 1972, he joined the "Simulator and Electronic Systems Division" of LE MATERIEL TELEPHONIQUE (LMT), and since 1977, he has been with the Sales Department where he is the leader for the LMT new products for Navies such as Ship Handling Simulators and Ship operations room Simulators.

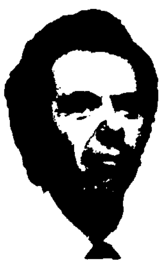
S. MATSUKI is a graduate of Kyushu University in 1948, and a professor of naval architecture at Kobe University of Mercantile Marine since 1973. He has taken part in research projects in a variety of marine-related affairs, mooring, waterway planning and so on.

J. B. MCHALE is a Chartered Engineer and Member of the Institution of Electronic and Radio Engineers. In 1967 he obtained H.N.C. with distinction in Electronic Engineering with endorsements in Control Engineering and Applied Electronics. He joined his present company, Y-ARD Ltd, in 1972 and has been primarily responsible for building up the companies capability and expertise in microprocessors, and is currently Head of the Advanced Control Technology and Instrumentation Section.

WILLIAM R. MCWHIRTER, JR., holds a B.S. Degree in Electrical Engineering from Virginia Polytechnic Institute and an M.S. Degree in Electrical Engineering (Systems Science) from The George Washington University. His experience has been in the area of instrumentation and control systems for shipboard machinery. He is currently a Project Engineer in the Automation, Control, and Systems Effectiveness Division at the David W. Taylor Naval Ship Research and Development Center, Annapolis, Maryland. He is a local chapter officer of the Institute of Electrical and Electronics Engineers and a member of the American Society of Naval Engineers.

DR. WALTER M. MACLEAN is presently the Manager, Requirements Development Laboratory of the National Maritime Research Center, Kings Point, New York. He holds a Doctor of Engineering in Naval Architecture from The University of California, Berkeley and is a Registered Mechanical Engineer in the State of California. His professional experience includes work in ship structure design of surface ships and submarines, ship structure research and the teaching of structural mechanics and naval architecture at Webb Institute of Naval Architecture and as Head, Department of Engineering, at the U.S. Merchant Marine Academy.

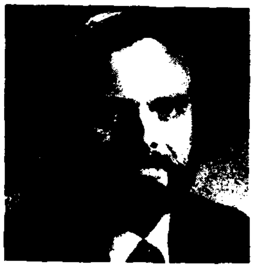
WARREN L. MALONE Head, Sensors and Controls Branch Surface Effect Ship Project (PMS 304) was graduated from the University of Illinois with a degree in Engineering Physics and received a Master's Degree in Physics from New York University. He has worked for approximately 20 years in the application and development of infrared, optical, and electromagnetic sensors. Since joining PMS 304 in 1972, he has been responsible for SES related technology in the areas of sensors, controls, and human factors. Mr. Malone holds patents for optical processing devices and has published numerous papers.



DAVID MANN received a Higher National Certificate in Electronic Engineering in 1964. From 1960 to 1968 he worked on the development of medium-sized central processor units for English Electric-Leo-Marconi Computers Ltd. In 1968 he joined Ultra Electronic Controls Ltd. and has since been concerned with the development of Electronic Control Systems for industrial and vehicle gas turbine engines. He is now Chief Project Engineer in the Vehicle Control Department.

THOMAS D. MARA currently manages a variety of marine research projects for the Maritime Administration, U.S. Coast Guard, and several ship operators. Previously he held positions as Managing Director of the CAORF ship simulator, Director of Product Development for Sperry Marine and other research positions in the marine field. He has been involved in the development of numerous advanced marine systems and integrated navigation systems since 1963.

NORMAN LINES joined Bristol Siddeley Engines in 1959 as an engineering apprentice and finally was awarded a BSc Hons degree in Mechanical Engineering in 1965. He joined the Fuel Systems group and has been employed on a number of control system projects involving the analogue simulation of closed loop control problems. He is currently involved in providing a simulation service to the Division and assessing equipment for future control systems.



CDR N. J. LOCKE won a scholarship to Royal Naval College, Dartmouth and later specialised in Marine Engineering at Royal Naval Engineering College, Plymouth. Served in cruisers as Marine Engineer Officer of frigate in charge of Machinery Controls Trials Team, attached to British Defence Liaison Staff, Ottawa and subsequently Naval Engineer Overseer, Yarrow Shipbuilders Ltd. Marine Engineer Officer of destroyer, HMS KENT and recently serving at Ministry of Defence Headquarters. He is a Chartered Engineer, Fellow Institute of Marine Engineers and Member Institution of Mechanical Engineers.

R. T. LOCOCK, BSc (Eng), MIEE, MINucE, C Eng Chief Constructor, Royal Corps of Naval Constructors (UK MOD (Navy)) has been since 1973 head of Design Sections at DG Ships, Bath responsible for Design and development of Surface Ship Main Electrical Power Supply Systems, Main Electrical Machinery and equipment, Electric Propulsion Systems and machinery (Patrol Submarine and Surface Ships). Since 1977 these responsibilities have also included Steering and Stabiliser Systems, Navigational Instruments and Lighting Equipment.

KEVIN P. LOGAN is an engineering analyst with Eclectech Associates, Inc. He is currently performing mathematical analyses and developing computer programs for application in the marine field. He has developed computer analysis techniques for the derivation of speed and fuel loss equations for merchant ships. He has participated in the experimental design and data processing of research in forced feedback systems linked to computer-driven control devices for submarine manipulators at Woods Hole Oceanographic Institution. Mr. Logan has a BA in Mathematics from Eastern Connecticut State College.

FREDDY KROGH was graduated from the University of Oslo in 1972 with a degree in Cybernetics. From 1972 to 1977 he performed computer simulation analysis of steam power plants, diesel engine systems, and ship maneuvering systems. He also participated in the development of computer systems for on-line supervision and control of engines onboard ship and the development cf general purpose simulator for control and supervision of dynamic systems.

D. E. LABBE holds a B.S. and M.S. degrees in Nuclear Engineering. Following three years as an engineer engaged in thermal-hydraulic development at the General Electric Co. Knolls Atomic Power Laboratory, Mr. Labbe is presently a Control System Engineer in the Medium Steam Turbine Department.

PROFESSOR T. H. LAMBERT is Professor of Automatic Control and Head of Department of Mechanical Engineering at University College London. Current research interests include an integrated ship control system to account for the multivariable nature of ship dynamics, submarine control systems and adaptive ships autopilots.

ANDRIES LAZET was born in The Hague, The Netherlands in 1920. After passing high school he went to the Royal Academy of Arts. He joined the Shell-Company for 10 years, and since 1954 he joined the Institute for Perception TNO. After a study-year in Canada he became head of the Human Engineering Research Department of the Institute.

K. KARASUNO received his masters and doctors degrees in Naval Architecture from Osaka University. He is an assistant professor of Kobe University of Mercantile Marine and a staff of ship handling simulator development project.

PETER R. KEYES is a Senior Program Manager at Mara-Time Marine Services, where he is responsible for various marine research projects. Included in his responsibilities are research programs being conducted at the Computer Aided Operations Research Facility (CAORF), Kings Point, New York. His prior experience includes employment by various shipping companies in engineering and operational capacities.

GENSHIRO KITAGAWA was born in Fukuoka, Japan, in 1948. He was graduated from the University of Tokyo, the Department of Mathematics and received B.S. and M.S. degrees in 1971 and 1973, respectively. Since 1974, has been with the Institute of Statistical Mathematics, Tokyo. His research interests are in time series analysis, system identification, general statistical model building and stochastic optimal control theory. He is a member of the Mathematical Society of Japan, the Japan Statistical Society, the society of Naval Architects and the Society of Instrument and Control Engineers.

TAKESHI KOJIMA graduated in 1963 from the Electronics Engineering School of Tohoku University. He has been employed by Mitsubishi Heavy Industries since 1963 and was assigned to Control Systems Engineering Group, Tokyo Head Office, and then to Nagasaki Shipyard & Engine Works No. 1 Ship Designing Department, Simulator Engineering Section as a senior engineer.

ROY D. JOHNSON received his B.E.E. from Cornell University in 1961 and his M.S. Degree in Electrical Engineering from Drexel University in 1965. His experience has been in the area of aerospace telemetry systems, experimental torpedo systems, speech analysis and synthesis, and control systems for shipboard machinery. He is currently working in the field of monitoring and fault detection systems for shipboard machinery at the David W. Taylor Naval Ship Research and Development Center, Annapolis, Maryland. He is a member of the Institute of Electrical and Electronics Engineers.

CLAES G. KÄLLSTRÖM received his M.S. degree (Civilingenjör) in electrical engineering from the Lund Institute of Technology, Lund, Sweden, in 1970. He is currently employed as a research engineer at the Department of Automatic Control at the Lund Institute of Technology. Since 1973 his research interests have been the application of modern control theory to ship dynamics. He is mainly interested in system identification and design of adaptive autopilots for ships.

JOHN G. KAMMERER is a Project Manager at the Naval Ocean Systems Center, San Diego. He received a BEE degree with honors from the University of Florida in 1959 and an MSEE from the University of Pennsylvania in 1963. Prior to joining NOSC in 1969, he held electronic system engineering positions with Computer Sciences Corporation, the Applied Physics Laboratory of Johns Hopkins University and RCA Missile and Surface Radar Division. He is a member of the IEEE, AFCEA and TAU BETA PI.

PAUL KAPLUN holds the B.S. (Physics) from City College of New York and M.S. (Fluid Dynamics) and D.Sc. (Applied Mechanics) from Stevens Institute of Technology. From 1951 to 1959 held several positions at the Davidson Laboratory of Stevens Institute of Technology. From 1959 to 1961 he was Chief of Hydrodynamics at TRG, Inc. Since 1961 he has been president of OCEANICS, Inc. Dr. Kaplan has acted as consultant on hydrodynamic problems to several organizations. He has published extensively in the area of hydrodynamics.

LT CDR JULIAN STEINHAUSEN gained an Honours Degree in Mechanical Engineering at the Royal Naval Engineering College Manadon. Having served in HMS NORFOLK for 2 years he returned to Manadon to complete the Advanced Marine Engineering Course in 1972. He served in HMS JUPITER and HMS INTREPID before taking up his current appointment in DG ships. He is currently working in the Machinery Controls section and is Project Officer for the Man/Machine Interface Design Study which is the subject of this paper.

A. M. STUURMAN received has degree in Naval Architecture from Delft University of Technology. Employed at the Directorate Materials of the Royal Netherlands Navy since 1969, he has been responsible for research related to the powering, maneuvering and control of both surface ships and submarines.

AKIRA SUGIMOTO graduated in 1971 from Control Engineering School of Tokyo Institute of Technology. He joined Mitsubishi Heavy Industries, Ltd. 1971, and has since been assigned to Nagasaki Technical Institute Control Research Laboratory to specialize in the research of ship control techniques.

JAMES R. E. THOMAS received the BSc. degree in Engineering (Electrical) from Leicester University, in 1973, and the MSc. degree, in Systems Engineering, from Surrey University in 1974. He is currently employed by Ferranti Computer Systems Ltd. working for the Systems Problems Analysis Group. He has been involved with the design and development of Naval Computer Systems and is currently engaged on a Satellite Radar System Study. Mr. Thomas is an associate member of the Institution of Electrical Engineers.

PROFESSOR R. V. THOMPSON head of Department of Marine Engineering at the University of Newcastle upon Tyne. Current interest include the development of improved combustion efficiency in power plant, dynamic analysis of geared systems and naturally propulsion system development, all of which he consults in on an International basis.

DR. ANTONIO TIANO graduated with a degree in Applied Mathematics from Genoa University. In 1973 he joined the "Laboratorio per l'Automazione Navale" as a scientific researcher and presently is mainly interested in the adaptive control of ships motions. He is a member of the Mathematical Society of Italy and of the Italian Group of Automatics and Systemistics Researchers and author of different papers concerning the application of identification and control techniques to the ship steering dynamics.

LT JAMES TONEY received his BSEE from Auburn University in 1972 and his MSME from the Naval Postgraduate School in September 1977. His duties have included tours as Communicator and Sonar Officer aboard the U.S. GRANT (SSBN 631). He is presently serving as Weapons Officer aboard the STONEWALL JACKSON (SSBN 634).

RONALD JOHN TURNER served as Navigating Officer, British Merchant Marine 1945-56. He found the staff of College of Nautical Studies in 1957 and since 1968, has served as Head of Navigation at the College in charge of the teaching and development of navigation. He is a Fellow of the Royal Institute of Navigation and author of numerous papers in Journal of Navigation and the maritime press.

TERRENCE GRAY TURNER was educated at Hurstpierpoint College and subsequently obtained a Full Technological Certificate in Telecommunications and Radio Engineering from the City and Guilds of London Institute. For the past four years with Vosper Thornycroft (UK) Limited Mr. Turner has been Project Manager for the Machinery Control System for the Royal Navy's new generation of Mine Counter Measures Vessels. During this period he has been closely associated with the initial development of the design and its subsequent introduction into full production.

JOB VAN AMERONGEN studied Electrotechnical Engineering at Delft University of Technology where he obtained his M.Sc. degree in 1971. Following military service with the Royal Netherlands Navy he joined the Control Group of the Electrical Engineering Dept. at Delft University of Technology as a scientific staffmember. His current research interests are, the applications of modern control algorithms in ship's steering and electric power plants, especially Model Reference Adaptive Control.

JAN VAN DAM studied at the University of Technology, Delft, where he received the M.S. degree in Applied Physiscs in 1951 and the Doctor's degree in Engineering in 1975. He joined the Royal Netherlands Naval College in 1964, where he is a Reader in Electronics and System Reliability now. His current interests are within application of Reliability Theory to naval problems. He is the author of several papers on this subject.

H. R. VAN NAUTA LEMKE graduated in Electrical Engineering at the Delft University of Technology in 1950. From 1950 - 1959 he worked in industry (Van der Heem, Philips) in the field of servo systems, instrumentation and sonar equipment. From 1959 he is a professor in control engineering at the Delft University. His interests are in the area of adaptive and optimal systems and the applications of fuzzy sets.

GUNNAR VERLO has been employed at Det norske Veritas Head Office since 1971 concerned with surveys of bridge systems and engine room automation, reliability analyses of bridge systems and risk analyses of marine operations. He is presently a part of a project team studying cause relationships of collisions and groundings.

PROFESSOR EZIO VOLTA obtained the degree of Doctor in Electrical Engineering from Genova University. He has been a full professor of Automatic Control at Genova University since 1964. He pioneered ships automation studies in Italy, and organized the first Italian research group working in the field. He is a member of the Associazione Elettrotecnica ed Elettronica Italiana. A member of the Institute of Electrical and Electronic Engineers (U.S.A.) and a member of several Italian and international Committees.

DR. JOHN R. WARE received Bachelor's and Master's in Mechanical Engineering from the University of Detroit and a PhD in Control Systems from the University of Michigan. From 1976 to the present, Dr. Ware has been employed by ORI, Inc., where he is primarily responsible for all aspects of ship control and submarine control. Dr. Ware's primary interests lie in applications of Optimal Control theory to advanced vehicles and modeling of human performance in manual control tasks.

DR. JOSEPH E. WHALEN received his BS degree in physics from Worcester Polytechnic Institute in 1966 and his PhD in physics from the University of Florida, Gainesville in 1972. He has been actively involved in simulating advanced Naval vehicles and sea environments for several years.

LCDR WHALLEY joined the Royal Navy from industry in 1964. He has served in HM Submarines on HMS ALBION and at the Royal Naval Engineering College, Plymouth, England. He holds the degrees of BSc., MSc., and PhD and is currently with DG Ships, Bath.

D.J. WHEELER, MSc C.Eng. joined Hawker Siddeley Aviation Ltd. as an apprentice in 1954 during which he trained at Coventry Technical College and then the College of Aeronautics. He continued with HSA Ltd. as a Development Engineer on Aircraft Systems and then in 1965 joined Bristol Siddeley later Rolls-Royce Ltd. to work on gas turbine control systems. He is now Engineer-in-Charge of Marine Systems.

HENRY K. WHITESEL received a Bachelor of Science degree in Engineering Science from Antioch College and a Master of Science degree in Electrical Engineering from George Washington University. A Senior Project Engineer at the David W. Taylor Naval Ship R&D Center he has designed, developed and evaluated shipboard instrumentation using lasers, nuclear magnetic resonance, vortex shedding, radar, acoustics and correlation techniques. He is presently interested in the development of shipboard instrumentation for monitoring and control applications.

PATRICK H. WHYTE obtained a B.A.Sc. in Aerospace Engineering from the University of Toronto in 1973, and an M.S.E. from Princeton University in 1975. His research interests involve the application of modern control theory to ship roll stabilization and control of the dynamics of advanced marine vehicles. He is a member of AIAA and an associate member of SNAME.

KENT E. WILLIAMS earned his PhD at the University of Connecticut. As a Senior Program Manager with Mara-Time Marine Services, Inc. he is responsible for coordination of efforts on research projects including a feasibility study on the inclusion of simulation in maritime training and licensing programs. Mr. Williams is a member of the American Psychological Association and Human Factors Society.

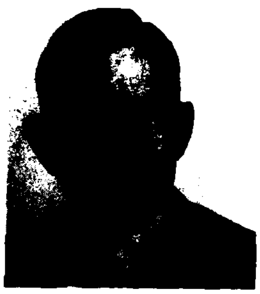
V.E. "BILL" WILLIAMS is located at Headquarters Support Laboratory, National Maritime Research Center, U.S. Merchant Marine Academy. Since joining the U.S. Maritime Administration he has been actively pursuing the field of in-service ship performance standards. Previously, he spent a number of years in industry in the ship control field.

JAMES C. WOLFORD received Bachelor's and Master's degrees in Electrical Engineering from Purdue University. He has seven years' experience at the Naval Weapons Support Center, Crane, Indiana, having served as project engineer for electronic designs of hybrid components, digital modules, and various military systems. Mr. Wolford is presently responsible for Crane's research and development for the U.S. Navy's Standard Electronic Module (SEM) Program.

JOHAN K. ZUIDWEG received an Ingenieur's degree in Electrical Engineering and a Doctor's degree in Engineering both from Delft Technological University. Since 1965 he has been teaching computer and control engineering at the Royal Netherlands Naval College, Den Helder, where he was appointed Reader, in 1969. Besides, he has been Visiting Lecturer and Reader respectively, in Electrical Engineering, at William Marsh Rice University, Houston, TX (1961-62), the University of Michigan, Ann Arbor, MI (1966-67) and the University of Nigeria, Nsukka (1973-74). His research interests are in various applications of systems theory.

SESSION CHAIRMAN BIOGRAPHIES

CAPT THOMAS L. ALBEE, JR., USN a graduate of the U.S. Naval Acadmey, also holds a Masters degree in Naval Architecture and Marine Engineering from Massachusetts Institute of Technology. He is a Submarine Qualified Engineering Duty Officer. Prior to his current assignment in the Naval Ship Engineering Center as Head of the Ship Systems Engineering and Design Department, he was Director of the Advance Technology Systems Division in the Naval Sea Systems Command. In his present assignment he supervises engineering and ship design for NAVSEC. He is a member of the National Council of the American Society of Naval Engineers and is currently the U.S. Project Officer on Information Exchange Project ABC-30 (Ship Control).



ROBERT C. ALLEN, was educated at Amherst College (A.B. Physics, 1948) and Yale University (M.S. Physics, 1949 and Ph.D. Physics, 1951). From 1951 to 1958 he was a staff member at Los Alamos Scientific Laboratory, Los Alamos, New Mexico, and from 1958 to 1967 he was Group Leader/Department Director at Atomics International, Canoga Park, California. Dr. Allen began his career at DTNSRDC in 1967 where he was Deputy Technical Director until 1970. He was Acting Technical Director of the Naval Ship R&D Laboratory, Panama City, Florida, from 1970-1971. From 1971 to date he has been Assistant Technical Director/Director of Technology, and since early 1978 Dr. Allen has been Acting Associate Technical Director for Propulsion and Auxiliary Systems and Acting Head, Propulsion and Auxiliary Systems Department.

CAPT A. TIMM ANDERSON, USN, holds a BS degree from the U.S. Naval Academy and an MS in Naval Architecture and Marine Engineering plus the Naval Engineers Degree from MIT. CAPT Anderson has served in surface ships, submarines and naval shipyards and as Chairman, Naval Systems Engineering Dept, USNA. Most recently he has been POLARIS/POSEIDON Submarine Project Officer, Submarine R&D Project Officer and Design Manager, NAVSEC and is presently Director, Machinery Systems Division, NAVSEC.

JAMES W. BANHAM is a graduate of the Pennsylvania State University in Mechanical Engineering. He currently holds positions both as Head of the Machinery Automation Systems Dept. of the Naval Ship Engineering Center, Philadelphia Div., and as Assistant Chairman of the Mechanical and Industrial Engineering Dept. of Drexel University's Evening College, where he holds the rank of Adjunct Associate Professor. He is the author of a text on Numerical Methods Applications in Engireering. In addition to numerous technical papers, he is also the author of the I^\ film on Boiler Feedwater Control Systems.

DR. WILLIAM E. CUMMINS holds the BS in Naval Architecture and Marine Engineering from Webb Institute and the PhD in Mathematics from American University. Since 1964 he has been Associate Technical Director and Head, Ship Performance Dept., David W. Taylor Naval Ship R&D Center. Dr. Cummins is a Fellow, Society of Naval Architects and Marine Engineers and of the Royal Institution of Naval Architects as well as a member of naval architectural societies of France and Japan.

WILLIAM J. DEJKA holds a BS degree in Electrical Engineering from the Illinois University and the MS in Engineering from U.C.L.A. Employed by the U.S. Navy since 1960, Mr. Dejka is presently a Senior Scientist, Naval Ocean Systems Center, where he has been developing long-range plans for the Navy in the areas of Distributed Systems and Computer Science. Mr. Dejka is a senior member of IEEE and Sigma Xi. He is the author of several professional papers and reports.

WILLIAM. M. ELLSWORTH is a graduate of the State University of Iowa where he obtained B.S. and M.S. degrees in Engineering, majoring in fluid mechanics. Since October 1969 he has been Associate Technical Director for Systems Development, and Head, Systems Development Department, Naval Ship Research and Development Center. He is a licensed Professional Engineer in the State of Maryland; a member of ASNE, SNAME, and ASME; and author of a number of papers and reports in the field of naval engineering. He served as a member of the ASNE Council from 1972 to 1974 and was nominated for Vice President in 1975. In 1974 he received the ASNE gold medal.

CAPTAIN (N) E. J. (ED) HEALEY is a graduate of the Canadian Military College Royal Roads and the Royal Naval Engineering College (RNEC) Manado at Plymouth, England. He served at sea, being engineering officer of three East Coast warships, interspersed with shore postings at the Fleet School and in the Dockyard. Following a year at Staff College, Captain Healey was posted to NAVSEC Philadelphia were he carried out the shore testing of the DDH 280 main machinery. Moving to Ottawa he became Design Section Head for Propulsion Machinery and Project Officer for the DDH 280 Propulsion Machinery. Following successful sea trials of the DDH 280 Class he was posted to the Esquimalt Naval Base as the Command Technical Officer. This was followed by a posting to the Canadian Forces Training Command and a year at the National Defence College. He is presently Director of Marine and Electrical Engineering at National Defence Headquarters.

DONALD H. KERN was educated at the Massachusetts Institute of Technology where he received his Bachelor of Science Degree in Naval Architecture and Marine Engineering and his Professional Engineers Degree in Naval Engineering. During his career in the U.S. Navy, he commanded the Portsmouth Naval Shipyard and served as Project and Design Manager for various submarine related programs. Currently Captain Kern is Vice President of Specialized Systems, Inc., of Mystic, CT.

REAR ADMIRAL JAMES W. LISANBY graduated from the U.S. Naval Academy and holds an advanced degree in Naval Engineering (Architecture) from the Massachusetts Institute of Technology. Rear Admiral Lisanby is an Engineering Duty Officer with wide and varied experience, both at sea and in shore assignments. From 1970 to 1973, he served as Supervisor of Shipbuilding at Pascagoula, Mississippi, with contract administration responsibilities for both the DD 963 and the LHA 1 ship acquisitions. Following a brief tour as Assistant for Ship Design in the Office of the Chief of Naval Operations, he served as Project Manager for the LHA Class of Amphibious Assault Ships with Headquarters in Washington, DC, from 1974 until he assumed his present position as Commander, Naval Ship Engineering Center, in 1978. Rear Admiral Lisanby holds the World War II Victory Medal, the Korean Service Medal with two battle stars, the United States Medal, and the National Defense Service Medal. In addition to these campaign ribbons, he has been awarded the Meritorious Service Medal for outstanding service with the Commander, Naval Ship Systems Command.

M. DALE MARTIN is a BSME graduate from West Virginia University. He began his government career as a project engineer in the Bureau of Ships, and through a series of progressively more responsible assignments, became head engineer for cargo systems with the Fast Deployment Logistic Ship Project, followed by head engineer for systems performance for the same project. Later he became Director of the NAVMAT Maintenance Technology Office, followed by Director of the Automatic Test Equipment Management and Technology Office. This office was recently reorganized into the present Test and Monitoring Systems Program Office, where Mr. Martin now serves as the Technical Director.

JOHN TODD McLANE obtained a B.S. degree in Physics at Lafayette College in 1951. He joined David W. Taylor Naval Ship Research and Development Center in 1963 and has studied human factors engineering at the Catholic University of America. While at the Center he has conducted psychophysical experiments in night vision, displays/controls, and color recognition. He has also studied the effects of ship motion on sailors and the human engineering design of Navy ship bridges. He is currently involved in the man/machine problems of shipboard machinery control monitoring systems. He is the Center's coordinator for human factors technology.

NILS H. NORRBIN was born in Nykoping, Sweden in 1926. He holds the Civilingenjor (Naval Architecture), Tekn. License and Tekn. Dr. From 1950 to 1954 he was with the Submarine Design and Preliminary Ship Design Departments of the Royal Swedish Naval Administration. Since 1955 he has been with the Research Department of the Swedish State Shipbuilding Experimental Tank as Head of the Ship Dynamics Section.

MICHAEL G. PARSONS, Associated Professor of Naval Architecture and Marine Engineering at the University of Michigan received a Ph.D. in Applied Mechanics specializing in optimal control from Stanford University in 1972. He returned to the University of Michigan in 1972 as Assistant Professor with teaching assignments in marine engineering, automatic control and computer-aided ship design. His current research interests include optimal control, optimization, and computer applications in naval architecture and marine engineering. Dr. Parsons is a member of Phi Eta Sigma, Tau Beta Pi, Phi Kappa Phi, Quarterdeck Society, and Scabbard and Blade Society.

ANTON CHARLES PIJCKE entered the Royal Netherlands Naval College at Den Helder (branch: Marine Engineering) in 1949 and received his commission as an officer in 1952. During his sea duty he served mostly onboard destroyers and frigates. He has been senior lecturer in Marine Engineering at the Royal Netherlands Naval College during several years and was also Head of the Naval Engineering Department. He left the Royal Netherlands Navy as a commander and is now a staff member of the Netherlands Maritime Institute at Rotterdam. Mr. Pijcke obtained his M.Sc. degree at London University, is a Fellow of the Institute of Marine Engineers, and is a chartered engineer. He is author of approximately twenty published papers. He was General Chairman of the 4th Ship Control Systems Symposium held at The Hague, The Netherlands in 1975.

CAPTAIN FREDERICK P. SCHUBERT, Deputy Chief, Office of Marine Environment and Systems, U.S. Coast Guard Headquarters, graduated from U.S. Coast Guard Academy in 1951 and received Masters degrees from Southern California and Auburn Universities. He has served 27 years in various operational assignments including duties in aviation and afloat as well as several staff assignments involving program management responsibilities. Captain Schubert assumed his present assignment in February of 1977. His duties include broad program management responsibilities in Port Safety and Security, Marine Environmental Protection, Aids to Navigation and Bridge Administration. Captain Schubert served as a member of the 1977 Department of Transportation Task Force on Marine Oil Transportation and Oil Pollution and was Chairman of the Pollution Response Working Group on the Interagency Task Force on Tanker Safety which developed the background for President Carter's Marine Oil Pollution Initiatives of 17 March 1977.

BRIAN SPENCER graduated in 1950 with an honours degree in mathematics. He became a member of the Scientific Advisers group in the Ship Department at Bath in 1965 with particular responsibility for automatic control systems and hydrodynamics. Has been very much involved with this series of Ship Control Systems Symposia, presented a paper at the First, organized the Third, a committee member for the Fourth, co-author of a paper to be presented at the Fifth, and U.K. coordinator.

RICHARD A. STANKEY received the B.S. degree from the Illinois Institute of Technology, Chicago, and a M.S. degree from Catholic University, Washington, DC, in Engineering Sciences and Mechanical Engineering, respectively. He came to the Naval Ship Engineering Center (NAVSEC) in Washington, DC in 1966 as a hydraulic system engineer. In 1970 he transferred to the Automatic Controls Section of the Ship Control Systems and Equipment Branch (Code 6165) of NAVSEC. He has done system analysis and design evaluation on hydrofoil flight control systems, submarine depth control and hovering systems, and surface ship position keeping and track keeping systems. He is currently working on the TRIDENT and Mine Countermeasure (MCM) Ship Control Systems.

JAMES STARK - Y-ARD Consultants, Ltd

Picture and biography not available at time of printing.

REAR ADMIRAL J. G. C. VAN DE LINDE began his higher education at the University of Delft in chemistry but was transported to Berlin to a forced labor camp in 1943. He escaped in 1944 and spent the rest of the war in the Dutch underground. In 1946 he went to the Royal Netherlands Naval College and was commissioned as an officer in 1948. He sailed in various submarines and surface ships and saw action in the Korean war. He specialized in torpedoes, weapon technology and nuclear reactor physics. After a four year command of the Royal Netherlands Naval College in Den Helder he became Chief of Naval Personnel in 1976.

H. R. VAN NAUTA LEMKE graduated in Electrical Engineering at the Delft University of Technology in 1950. From 1950 - 1959 he worked in industry (Van der Heem, Philips) in the field of servo systems, instrumentation and sonar equipment. From 1959 he is a professor in control engineering at the Delft University. His interests are in the area of adaptive and optimal systems and the applications of fuzzy sets.

LCDR IR. WILLEM VERHAGE was born in Rotterdam in 1940. After completing high school he enrolled in the school for merchant navy in Amsterdam. He joined the Royal Dutch Navy in 1961 and served as an instructor at the Naval Communications School in Amsterdam. In 1969 he was granted his engineering degree in naval architecture from Delft University of Technology. Since 1970 Ir. Verhage serves at the RNNC in Den Helder where he holds the rank of LCDR. In addition to his teaching duties in the department of naval architecture he is head of a research team which developed a nocturnal simulator.

STANLEY D. WHEATLEY - Maritime Administration

Picture and biography not available at time of printing.

SYMPOSIUM GUEST SPEAKERS

The Honorable David E. Mann

Keynote Address

"Ship System Integration for Future Design"

Randolph W. King

Dinner-Guest Speaker

"The Implication of New Technology for the Maritime Community"

FIFTH SHIP CONTROL SYSTEMS SYMPOSIUM

DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Captain Myron V. Ricketts
Commander

Dr. Alan Powell
Technical Director

Dr. Robert C. Allen
Head, Propulsion and Auxiliary
Systems Department

Mr. Walter J. Blumberg
Head, Automation, Control, and
Systems Effectiveness Division

Symposium Organization

Walter J. Blumberg
General Chairman

Michael A. Gawitt
Technical Chairman

Charles C. Hall, Jr.
Publications Chairman

Edwin M. Petrisko
Registration Chairman

Margaret C. Paxson
Administration

E. Brent Richardson
Arrangements

DAT

ILM